Heating Ventilating
Air Conditioning
Guide 1947

HEATING VENTILATING AIR CONDITIONING GUIDE

1947

AN INSTRUMENT OF SERVICE PREPARED FOR THE PROFESSION—CONTAINING A

Technical Data Section

OF REFERENCE MATERIAL ON THE DESIGN AND SPECIFICATION OF HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS BASED ON—THE TRANSACTIONS—THE INVESTIGATIONS OF THE RESEARCH LABORATORY AND CO-OPERATING INSTITUTIONS—AND THE PRACTICE OF THE MEMBERS AND FRIENDS OF THE SOCIETY

TOGETHER WITH A

Manufacturers' Catalog Data Section

CONTAINING ESSENTIAL AND RELIABLE INFORMATION CONCERNING MODERN EQUIPMENT

ALSO

The Roll of Membership of the Society

WITH

Complete Indexes

TO TECHNICAL AND CATALOG DATA SECTIONS

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PREFACE TO THE 25th ANNIVERSARY EDITION

THIS 25th Anniversary Edition of THE HEATING, VENTILATING, AIR CONDITIONING GUIDE is the culmination of a quarter century of effort by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in serving its members and the allied profession and industry.

In the Silver Anniversary Edition, THE GUIDE 1947 presents an expanded Technical Data Section of 912 pages covering theory and current practice in the fields of heating, ventilating, and air conditioning. Wherever possible, previous text has been improved and condensed to provide additional space for new material. All technical data have been carefully reviewed and the same order of chapters which appeared previously has been retained. The chapters which received the most critical study and revision in this 1947 edition are noted in the following paragraphs:

Chapter 1, Terminology. Many of the terms have been redefined to attain accuracy and clarity. Definitions of several useful terms have been added.

Chapter 3, Thermodynamics, has been enlarged by the addition of a steam table covering the pressure range from atmospheric to 500 psia.

Chapter 7, Performance of Air Heating and Cooling Coils, is limited to fundamentals determining performance of coil surface. Practical factors relating to coil selection have been transferred to Chapter 25.

Chapter 12, Physiological Principles, has been edited to include data from past and present Society research, as well as information from recent investigations conducted by the armed services on the effect of environment upon human comfort and endurance.

Chapter 13, Air Conditioning in the Prevention and Treatment of Disease, has been subject to certain revisions which keep it in accord with current practice

In *Chapter 14*, Heating Load, the number of cities for which climatological data are given has been greatly enlarged. Bands of design temperature, based upon the information given for the various cities, are shown on a new outline map of the United States.

Chapter 15, Cooling Load, has been completely rewritten and includes an enlarged list of cities for which summer design conditions are shown. Methods of determining components of heat and water vapor gain are given in detail for homogeneous and composite construction. A detailed example in determining cooling load is included.

Chapter 16, Fuels and Combustion, has been enlarged and rearranged for more effective presentation of the subject. Formulas for heat losses have been added and several useful charts are included for determining flue gas losses.

Much of the text and some formulas have been amplified to increase the usefulness of *Chapter 20*, Estimating Fuel Consumption for Space Heating.

Chapter 24, Hot Water Heating Systems and Piping, has been revised in order to simplify the examples in system design. A formula has been added for determining the size of expansion tank in a closed system.

Chapter 25, Radiators, Convectors, Coils, has been greatly expanded by inclusion of information on practical factors affecting the selection of coils formerly appearing in Chapter 7.

Chapter 33, Air Cleaning Devices, has been considerably revised and includes additional information of particular value for the user of air cleaners. A section has been added to cover the subject Methods of Installation.

Chapter 36, Unit Air Conditioners, Unit Air Coolers, Attic Fans, has been largely rewritten and contains improved sections referring to the features of construction and factors influencing operation and application of unit equipment.

Chapter 38, Dehumidification by Sorbent Materials, has been rewritten to conform to present practice and knowledge in the sorbents field. An illustrative example, showing a method of calculating the moisture load, is included

Chapter 39, Refrigeration, includes a new section describing the Air Refrigeration Cycle. The section referring to Absorption Systems has been simplified.

The friction loss chart in Chapter 41, Air Duct Design, has been extended to include ducts of small size to a minimum of 1½ in. diameter. A chart for facilitating correction of friction loss for pipe roughness has been added.

Chapter 49, Marine Heating, Ventilation, Air Conditioning, has been revised thoroughly and includes new material based upon both Navy and merchant marine practice in air conditioning and dehumidification. Typical air conditioning systems for passenger ships are shown diagrammatically.

In response to many requests for more detailed data, *Chapter 50*, Water Services, has been improved by the addition of information on design of cold water piping systems. A chart for determining friction loss in rough pipe has been added to the previous chart showing friction loss for fairly rough pipe.

Many codes and standards, which were published during the past year, have been added to *Chapter 51*, Codes and Standards, to provide an up-to-date and useful list of codes for ready reference.

A cross index of the Technical Data Section is included immediately following the Table of Contents.

The progress made by The Guide during its 25 years of service and its ability to reflect current practice, results from the generous cooperation of many Society members, organizations, and individual authorities, whose public spirited efforts are rewarded only by the satisfaction derived from performing a useful service to the profession and the public. We salute those farseeing members who envisioned The Guide and who guided it through its formative years. We pay tribute to the work of the former Guide Publication Committees and those contributors who pioneered in establishing this reference data book, which, during its quarter century of service, has received world wide recognition and acclaim. To that brilliant group who made The Guide outstanding in previous years, we take great pleasure in adding the following names in recognition of their faithful service and valuable contributions to this 25th edition:

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Among the 241 firms represented in the Catalog Data Section there are nine leading manufacturers whose catalog material has appeared in every edition since 1922. The data on the modern equipment described, have been carefully edited in an effort to eliminate exaggerated claims or statements and should prove to be a helpful supplement to the technical data.

The Committee releases this Silver Anniversary Edition with the sincere hope that it will be a worthy successor to the 24 preceding editions, in advancing the arts of Heating, Ventilating, and Air Conditioning.

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CODE of ETHICS for ENGINEERS

ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6—He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.

7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

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CHAPTER 1

Terminology

Glossary of Physical and Heating, Ventilating, Refrigerating and Air Conditioning Terms Used in the Text

Absolute Zero: The zero from which absolute temperature is reckoned. Approximately −273.2 C or −459 8 F.

Absorbent: A sorbent which changes physically or chemically, or both, during the sorption process

Absorption: The action of a material in extracting one or more substances present in an atmosphere or mixture of gases or liquids; accompanied by physical change, chemical change, or both, of the sorbent.

Acceleration: The time rate of change of velocity *i e.*, the derivative of velocity with respect to time. In the cgs system the unit of acceleration is the centimeter per (second) (second), in the fps system the unit is the foot per (second) (second) $a = \frac{dv}{dt}$.

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body, the value of which varies with latitude and elevation. The international gravity standard has the value of 980.665 cm per (second) (second) or 32 174 ft per (second) (second) which is the actual value of this acceleration at sea level and about 45 deg latitude

Adiabatic: An adjective descriptive of a process such that no heat is added to or taken from a substance or system undergoing the process.

Adsorbent: A sorbent which does not change physically or chemically during the sorption process.

Adsorption: The action, associated with surface adherence, of a material in extracting one or more substances present in an atmosphere or mixture of gases and liquids unaccompanied by physical or chemical change. Commercial adsorbent materials have enormous internal surfaces

Aerosol: An assemblage of small particles, solid or liquid, suspended in air. The diameters of the particles may vary from 100 microns down to 0 01 micron or less, *e.g.* dust, fog, smoke.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, gases, vapors, fumes and smokes. (Air cleaners include air washers, air filters, electrostatic precipitators and charcoal filters.)

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure These factors include temperature, humidity, motion, distribution, dust, bacteria, odors and toxic gases, most of which affect in greater or lesser degree human health or comfort. (*See Comfort Air Conditioning.*)

Air, Dry: In psychrometry, air unmixed with, or containing no, water vapor.

Air, Saturated: A mixture of dry air and saturated water vapor, all at the same dry-bulb temperature.

Air, Standard: Air with a density of 0.075 lb per cubic foot and an absolute viscosity of 1.22×10^{-5} lb mass per (foot) (second). This is substantially equivalent to dry air at 70 F and 29 92 in. (Hg) barometer.

Air Washer: An enclosure in which air is drawn or forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of a fluid.

Aspect Ratio: In air distribution outlets the ratio of the length of the core of a grille, face or register to the width.

In rectangular ducts the ratio of the width to the depth.

Atmospheric Pressure: The pressure due to the weight of the atmosphere. It is the pressure indicated by a barometer. *Standard Atmospheric Pressure* or *Standard Atmosphere* is the pressure of 76 cm of mercury having a density of 13 5951 grams per cubic centimeter, under standard gravity of 980.665 cm per (second) (second). It is equivalent to 14.696 lb per square inch or 29.921 in. of mercury at 32 F.

Baffle: A surface used for deflecting fluids, usually in the form of a plate or wall.

Blast Heater: A set of heat transfer coils or sections used to heat air which is drawn or forced through it by a fan.

Blow (throw): In air distribution, the distance an air stream travels from an outlet to a position at which air motion along the axis reduces to a velocity of 50 fpm.

For unit heaters, the distance an air stream travels from a heater without a perceptible rise due to temperature difference and loss of velocity.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929.)

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hour from and at 212 F. This is equal to a heat output of 970.3 × 34.5 = 33,475 Btu per hour.

British Thermal Unit: Classically the Btu is defined as the quantity of heat required to raise the temperature of 1 lb of water 1 Fahrenheit degree. By this definition the exact value depends upon the initial temperature of the water. Several values of the Btu are in more or less common use, each differing from the others by a slight amount. One of the more common of these is the *mean Btu* which is defined as 1/180 of the heat required to raise the temperature of 1 lb of water from 32 F to 212 F at a constant atmospheric pressure of 14.696 lb per square inch absolute.

For most accurate work the *International Table* (I.T.) Btu is usually used. This is defined by the relation: 1 (I.T.) Btu per (pound) (Fahrenheit degree) = 1 (I.T.) calorie per (gram) (Centigrade degree). This value corresponds to the amount of heat required to raise the temperature of 1 lb of water 1 Fahrenheit degree at 58 F and also at 149 F. The *mean Btu* corresponds to 1.0008 (I.T.) Btu.

By-Pass: A pipe or duct, usually controlled by valve or damper, for conveying a fluid around an element of a system.

Calorie (Gram Calorie): Classically the calorie is defined as the quantity of heat required to raise the temperature of 1 gram of water 1 Centigrade degree. By this definition the exact value depends upon the initial temperature of the water. Several values of the calorie are in more or less common use, each differing from the others by a slight amount. Among these are the *15 C calorie* and the *17½ C calorie*. The *mean calorie*, *i.e.* 1/100 the quantity of heat required to raise the temperature of 1 gram of water from 0 C to 100 C, is also extensively used.

For the most accurate work the International Table (I.T.) calorie, defined in terms of the international electrical units, is usually used: 1 (I.T.) calorie = 1/860 international watt-hour = 3,600/860 international watt-seconds or international joules.

The kilocalorie = 1,000 cal.

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and is conveyed to and from the rooms by means of a fan and a system of distributing ducts. (See Chapter 43.)

Chimney Effect: The tendency of air or gas in a duct or other vertical passage to rise when heated due to its lower density compared with that of the surrounding air or gas. In buildings, the tendency toward displacement (caused by the difference in temperature) of internal heated air by unheated outside air due to the difference in density of outside and inside air.

Comfort Air Conditioning: The process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. (See *Air Conditioning.*)

Comfort Line: The effective temperature at which the largest percentage of adults feels comfortable.

Comfort Zone (*Average*)**:** The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. (See Chapter 12.)

Condensate: The liquid formed by condensation of a vapor. In steam heating, water condensed from steam; in air conditioning, water extracted from air, as by condensation on the cooling coil of a refrigeration machine.

Condensation: The process of changing a vapor into liquid by the extraction of heat. Condensation of steam or water vapor is effected in either steam condensers or in dehumidifying coils and the resulting water is called *condensate*.

Conductance, Surface (*Unit*)**:** The amount of heat transferred by radiation, conduction, and convection from unit area of a surface to the air or other fluid in contact with it, or vice versa, in unit time for a unit difference in temperature between the surface and the fluid. The common unit is: Btu per (hour) (square foot) (Fahrenheit degree). Symbol *f*. The temperature of the fluid should be taken in a plane sufficiently ... ill not be affected by the temperature of the surface.

Conductance, Thermal: The time rate of heat flow through unit area of a body, of given size and shape, per unit temperature difference. Common unit is: Btu per (hour) (square foot) (Fahrenheit degree). Symbol *C*.

Conduction, Thermal: The process of heat transfer through a material medium in which kinetic energy is transmitted by the particles of the material from particle to particle without gross displacement of the particles.

Conductivity, Thermal: The time rate of heat flow through unit area of a homogeneous substance under the influence of a unit temperature gradient. Common units are: Btu per (hour) (square foot) (Fahrenheit degree per inch). Symbol *k*.

Conductor, Thermal: A material which readily transmits heat by means of conduction.

Convection: The motion resulting in a fluid from the differences in density and the action of gravity. In heat transmission this meaning has been extended to include both *forced* and *natural* motion or circulation.

Convective Heat Transfer: The transmission of heat by either natural or forced motion of a fluid (liquid or gas).

Convector: An agency of convection. In heat transfer, a surface designed to transfer its heat to a surrounding fluid largely or wholly by convection. The heated fluid may be removed mechanically or by gravity (Gravity Convector). Such a surface may or may not be enclosed or concealed. When concealed and enclosed the resulting device is sometimes referred to as a concealed radiator. (See also definition of *Radiator*.) (See also Chapter 25.)

Decibel: A unit used to express the relation between two amounts of power. By definition the difference in decibels between two powers P_1 and P_2, P_2 being the larger, is

$$\text{db difference} = 10 \log_{10} \frac{P_2}{P_1}$$

In acoustics the threshold of hearing at 1,000 cycles per second has been standardized at 10^{-16} watts per square centimeter. If P_2 is the power in watts per square centimeter of a measured sound, then $10 \log_{10} \frac{P_2}{10^{-16}}$ is the db difference above the threshold and is known as the *intensity level*. This is a definite recognized way of describing the intensity of a sound.

Degree-Day: A unit, based upon temperature difference and time, used in estimating fuel consumption and specifying nominal heating load of a building in winter. For any one day, when the mean temperature is less than 65 F, there exists as many degree-days as there are Fahrenheit degrees difference in temperature between the mean temperature for the day and 65 F.

Dehumidify: To reduce, by any process, the quantity of water vapor within a given space.

Dehydrate: To remove water in all forms from matter. Liquid water, hygroscopic water, and water of crystallization or water of hydration are included.

Density: The ratio of the mass of a specimen of a substance to the volume of the specimen. The mass of a unit volume of a substance. When weight can be used without confusion, as synonymous with mass, density is the weight of a unit volume of a substance.

Dew-Point: See *Temperature, Dew-Point*.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct-Return System (*Hot Water*)**:** A hot water system in which the water, after it has passed through a heating unit, is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser (*Steam*)**:** A pipe which carries steam downward to the heating units and into which the condensate from the heating units drains.

Down-Feed System (*Steam*)**:** A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft: A current of air. Usually refers to the pressure difference which causes a current of air or gases to flow through a flue, chimney, heater or space.

Draft Head (*Side Outlet Enclosure*)**:** The height of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening. (*Top Outlet Enclosure*)**:** The height of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Drip: A pipe, or a steam trap and a pipe considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Dry: To separate or remove a liquid or vapor from another substance. The liquid may be water but the term is also used for the removal of liquid or vapor forms of other substances.

Dust: An air suspension (aerosol) of solid particles of any material. (See also Chapter 10, p. 170.)

Enthalpy: A term used in lieu of *total heat* or *heat content*. Expressible in Btu per pound. Mathematically defined as $h = u + pv / J$. When a change occurs at constant pressure, as when water is boiled, the change in enthalpy is equal to the heat added, in this case latent heat.

Enthalpy, Free: A thermodynamic property which serves as a measure of the available energy of a system with respect to surroundings at the same temperature and same pressure as that of the system. No process involving an increase in available energy can occur spontaneously.

Enthalpy, Specific: A term sometimes applied to enthalpy per unit weight, the English unit being Btu per pound.

Entropy: The ratio of the heat added to a substance to the absolute temperature at which it is added. Mathematically, for a reversible process,

$$dS = \frac{dQ}{T} \text{ or } S = \int \frac{dQ}{T}$$

These formulas are applicable when temperature is not constant. During a reversible adiabatic change entropy is constant. During a reversible isothermal change the heat absorbed by the substance is equal to the product of the absolute temperature of the substance and its change of entropy.

Entropy, Specific: A term sometimes applied to entropy per unit weight, the English unit being Btu per (Fahrenheit degree, absolute) (pound).

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and corresponding atmospheric pressure.

Fan Furnace System: See *Warm Air Heating System.*

Fog: Suspended liquid droplets generated by condensation from the gaseous to the liquid state or by breaking up a liquid into a dispersed state, such as by splashing, *foaming*, and *atomizing*. (See also Chapter 10, p. 171.)

Force: The action on a body which tends to change its relative condition as to rest or motion.

Fumes: Smoke; aromatic smoke; odor emitted, as of flowers; a smoky or vaporous exhalation, usually odorous, as that from concentrated nitric acid. The word *fumes* is so broad and inclusive that its usefulness as a technical term is very limited. Its principal definitive characteristic is that it implies an odor. The terms vapor, smoke, fog, etc., which can be more strictly defined, should be used whenever possible.

Also defined as solid particles generated by condensation from the gaseous state, generally after volatilization from molten metals, etc., and often accompanied by a chemical reaction such as oxidation. Fumes flocculate and sometimes coalesce. (See also Chapter 10, p. 170.)

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also a complete heating unit for transferring heat from fuel being burned to the air supplied to a heating system.

Furnace Volume (*Total*)**:** The total furnace volume for horizontal-return tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal-return tubular boiler settings, unless manifestly ineffective (*i.e*, no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes. (*A.S.M.E.* Power Test Codes, Series 1929.)

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down draft boilers,

the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as previously defined.

Gravity, Specific: The ratio of the mass of a unit volume of a substance to the mass of the same volume of a standard substance at a standard temperature. Water at 39.2 F is the standard substance usually referred to. For gases, dry air, at the same temperature and pressure as the gas, is often taken as the standard substance.

Gravity Warm Air Heating System: See *Warm Air Heating System.*

Head, Dynamic: Same as Total Pressure expressed in height of liquid.

Heat: The form of energy that is transferred by virtue of a temperature difference. At constant pressure heat added is equal to enthalpy change.

Heat, Humid: Ratio of increase of enthalpy per pound of dry air to rise of temperature under conditions of constant pressure and constant humidity ratio.

Heat, Latent: A term used to express the energy involved in a change of state.

Heat, Sensible: A term used in heating and cooling to indicate any portion of heat which changes only the temperature of the substances involved.

Heat of the Liquid: The increase in enthalpy per unit weight of a saturated liquid as its temperature increases from a chosen base temperature. For water the base temperature is usually taken as 32 F.

Heat, Specific: The heat absorbed (or given up) by a unit mass of a substance when its temperature is increased (or decreased) by 1 deg. Common Units: Btu per (pound) (Fahrenheit degree), calories per (gram) (Centigrade degree). For gases, both specific heat at constant pressure (C_p) and specific heat at constant volume (C_v) are frequently used. In air-conditioning, C_p is usually used.

Heat, Total: See Enthalpy.

Heat Transmission, Coefficient: Any one of a number of coefficients used in the calculation of heat transmission by conduction, convection, and radiation, through various materials and structures. (See thermal conductance, thermal conductivity, thermal resistance, thermal resistivity, thermal transmittance, etc.).

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried from the boiler to the heating units.

Humidify: To increase, by any process, the density of water vapor within a given space.

Humidistat: A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

Humidity: Water vapor within a given space.

Humidity, Absolute: The weight of water vapor per unit volume, pounds per cubic foot or grams per cubic centimeter.

Humidity, Relative: The ratio of the actual partial pressure of the water vapor in a space to the saturation pressure of pure water at the same temperature. (See discussion in Chapter 3.)

Humidity Ratio: In a mixture of water vapor and air, the weight of water vapor per pound of dry air. Also called *Specific Humidity.*

Humidity, Specific: See Humidity Ratio.

Hygrostat: Same as Humidistat.

Inch of Water: A unit of pressure equal to the pressure exerted by a column of liquid water 1 in. high at a standard temperature. The standard temperature is sometimes taken as 0 C and sometimes as 62 F. One inch of water at 62 F = 5.197 lb per square foot.

Insulation (*Thermal*)**:** A material having a relatively high resistance to heat flow, and used principally to retard the flow of heat.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Load, Estimated Design: In a heating or cooling system, the sum of the useful heat transfer plus heat transfer from or to the connected piping plus heat transfer occurring in any auxiliary apparatus connected to the system. The units are Btu per hour or, in heating, equivalent direct radiation (EDR). The unit EDR is becoming obsolete.

Load, Estimated Maximum: In a heating or cooling system, the calculated maximum heat transfer that the system will be called upon to provide.

Manometer: An instrument for measuring pressures; essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so constructed that the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: A measure of the inertia of a body. It also measures the quantity of matter in a body. Since the only general property of a given portion of matter that cannot be changed is its inertia, it is this property by which quantities of matter are defined. Two bodies which have equal inertias are said to have equal masses or to contain equal quantities of matter. (This definition fails at velocities approaching the velocity of light.) The mass of a body is numerically equal to the ratio of the force required to give the body a given acceleration to the acceleration. $m = \frac{F}{a}$. The common units of mass are the gram and the pound.

Mechanical Equivalent of Heat: The quantity of mechanical energy equal to one unit of heat. $J = 778.3$ ft-lb per Btu $= 4.187 \times 10^7$ ergs per gram-calorie.

Medium, Heating: A substance such as water, steam, air or furnace gas used to convey heat from the boiler, furnace or other souce of heat or energy to the heating unit from which the heat is dissipated.

Micron: A unit of length, the thousandth part of 1 mm or the millionth of a meter.

Millimeter of Mercury: A unit of pressure equal to the pressure exerted by a column of mercury 1 mm high at a temperature of 0 C. One millimeter of mercury at 0 C $= 1.934 \times 10^{-2}$ lb per square inch.

Mol: A weight of a substance numerically equal to its molecular weight. If the weight is in pounds the unit is a *Pound Mol*, in grams the unit is a *Gram Mol*. For perfect gases the volume of 1 mol is constant for all gases at the same temperature and pressure. For real gases this is approximately true at moderate pressures. At 32 F and zero-pressure the value of the product, pressure times specific volume, is 359.045 ± 0.006 atmosphere cubic feet (atm ft^3), *for 1 mol of any gas*. For dry air at 32 F and standard atmospheric pressure, the specific volume is 358.83 cu ft per mol (ft^3 per mol).

One-Pipe Supply Riser—(*Steam*): A pipe which carries steam vertically to a heating unit and which also carries the condensate from the heating unit. In an upfeed system steam and condensate flow in opposite directions; in an overhead or down-feed system they flow in the same direction.

One-Pipe System—(*Steam*): A steam heating system in which a single main serves the dual purpose of supplying steam to the heating unit and conveying condensate from it. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used. (Hot Water)—A hot water system in which the cooled water from the heating units is returned to the supply main. Consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

Overhead System: Any steam or hot water system in which the supply main is above the heating unit. In a steam system the return must be below the heating units; in a water system the return may be above or below the heating units.

Panel Heating: A heating system in which heat is transmitted by both radiation and convection from panel surfaces to both air and surrounding surfaces.

Panel Radiator: A heating unit placed on or flush with a flat wall surface and intended to function essentially as a radiator.

Plenum Chamber: An air compartment maintained under pressure and connected to one or more distributing ducts.

Potentiometer: An instrument for comparing small electromotive forces or for measuring small electromotive forces by comparison with a known electromotive force. Its principal advantage is that during the measurement no current flows through the source of electromotive force.

Power: The rate of performing work. Common units are horsepower, Btu per hour, and watts.

Pressure: Force per unit area. Common units are pounds per square inch, gram per square centimeter, inch of water, millimeter of mercury.

Pressure, Absolute: The sum of the gage pressure and the barometric pressure.

Pressure, Gage: Pressure measured from atmospheric pressure as a base. Gage pressure may be indicated by a manometer which has one leg connected to the pressure source and the other exposed to atmospheric pressure.

Pressure, Dynamic: Same as Total Pressure.

Pressure, Saturation: The saturation pressure for a pure substance for any given temperature is that pressure at which vapor and liquid or vapor and solid can co-exist in stable equilibrium.

Pressure, Static: The normal force per unit area that would be exerted by a moving fluid on a small body immersed in it if the body were carried along with the fluid. Practically, it is the normal force per unit area at a small hole in a wall of the duct through which the fluid flows (piezometer) or on the surface of a stationary tube at a point where the disturbances created by inserting the tube cancel. It is supposed that the thermodynamic properties of a moving fluid depend on static pressure in exactly the same manner as those of the same fluid at rest depend upon its uniform hydrostatic pressure.

Pressure, Total: In the theory of the flow of fluids; the sum of the static pressure and the velocity pressure at the point of measurement.

Pressure, Vapor: The pressure exerted by a vapor. If a vapor is kept in confinement over its liquid so that the vapor can accumulate above the liquid, the temperature being held constant, the vapor pressure approaches a fixed limit called the maximum, or saturated, vapor pressure, dependent only on the temperature and the liquid. The term *vapor pressure* is sometimes used as synonymous with *saturated vapor pressure.*

Pressure, Velocity: In a moving fluid, the pressure capable of causing an equivalent velocity if applied to move the same fluid through an orifice such that all pressure energy expended is converted into kinetic energy.

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere.

Psychrometric: Pertaining to psychrometry or the state of the atmosphere with reference to moisture.

Psychrometry: The branch of physics relating to the measurement or determination of atmospheric conditions, particularly regarding the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiant Heating: A heating system in which only the heat radiated from panels is effective in providing the heating requirements. The term *Radiant Heating* is frequently used to include both *Panel and Radiant Heating.*

Radiation: The transmission of energy by means of electromagnetic waves.

Radiation, Thermal (Heat) Radiation: The transmission of energy by means of electromagnetic waves of very long wave length. Radiant energy of any wave length may, when absorbed, become thermal energy and result in an increase in the temperature of the absorbing body.

Radiation, Equivalent Direct (*EDR, Steam*)**:** That amount of heating surface expressed in square feet, which will deliver 240 Btu per hour, under the design operating conditions. (*EDR, Hot Water*)**:** That amount of heating surface, expressed in square feet, which will deliver 150 Btu per hour, under the design operating conditions. Thus, 1 sq ft of EDR does not imply 144 sq in. of heater surface, but means a heat delivery of 240 (or 150) Btu per hour for each EDR of a given radiator or convector.

Radiator: A heating unit exposed to view within the room or space to be heated. A radiator transfers heat by radiation to objects within visible range and by conduction to the surrounding air which in turn is circulated by natural convection; a so-called radiator is also a convector but the single term radiator has been established by long usage.

Radiator, Concealed: A heating device located within, adjacent to, or exterior to the room being heated but so covered or enclosed or concealed that the heat transfer surface of the device, which may be either a radiator or a convector, is not visible from the room. Such a device transfers its heat to the room largely by convection air currents.

Radiator, Direct: Same as Radiator.

Radiator, Recessed: A heating unit set back into a wall recess but not enclosed

Radiator, Tube or Tubular: A heating unit used as a radiator in which the heat transfer surfaces are principally tubes.

Refrigerant: A substance which produces a refrigerating effect by its absorption of heat while expanding or vaporizing.

Refrigeration, Ton of: The removal of heat at a rate of 200 Btu per minute, 12,000 Btu per hour, or 288,000 Btu per 24 hours.

Resistance, Thermal: The reciprocal of thermal conductance. Symbol R.

Resistivity, Thermal: The reciprocal of thermal conductivity. Symbol r.

Return, Dry: A return pipe in a steam heating system which carries both water of condensation and air. The dry return is above the level of the water line in the boiler in a gravity system. (See *Return, Wet.*)

Return, Wet: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so. (See *Return, Dry.*)

Return Mains: Pipes or conduits which return the heating or cooling medium from the heat transfer unit to the source of heat or refrigeration.

Reversed-Return System: A system in which the heating or cooling medium from several heat transfer units is returned along paths arranged so that all circuits composing the system or composing a major sub-division of the system are of practically equal length.

Saturation: The condition for co-existence in stable equilibrium of a vapor and liquid or a vapor and solid phase of the same substance. Example: Steam over the water from which it is being generated.

Saturation, Degree of, or Per Cent: The ratio of the weight of a given volume of water vapor to the weight of an equal volume of saturated water vapor at the same temperature.

Smoke: An air suspension (aerosol) of particles, usually but not necessarily solid, often originating in a solid nucleus, formed from combustion or sublimation. Also defined as carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion and thereby to reduce the smoke produced.

Solar Constant: The solar intensity incident on a normal surface located outside the earth's atmosphere at a distance from the sun equal to the mean distance between the earth and the sun. Its value is 415, 445, or 430 Btu per (hour) (square foot) as the July, January, or mean value respectively. At sea level in July the solar intensity value is about 300 Btu per (square foot) (hour) since about 28 per cent is absorbed in the earth's atmosphere.

Sorbent: A material which extracts one or more substances present in an atmosphere or mixture of gases or liquids with which it is in contact, due to an affinity for such substances.

Sorption: Adsorption or absorption.

Split System: A system in which the heating is accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point. Ventilation may be provided by the same system.

Square Foot of Heating Surface (*Equivalent*)**:** This term is synonymous with Equivalent Direct Radiation (EDR).

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Steam: Water in the vapor phase. *Dry Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing no water in suspension. *Wet Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing water particles in suspension. *Superheated Steam* is steam at a temperature higher than the saturation temperature corresponding to the pressure.

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of heat to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate, or of air and condensate and preventing the passage of steam.

Supply Mains: The pipes through which the heating medium flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface, Heating: The exterior surface of a heating unit. *Extended heating surface* (*or extended surface*): Heating surface consisting of fins, pins or ribs which receive heat by conduction from the prime surface. *Prime Surface:* Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also Boiler Heating Surface.)

Temperature: The thermal state of matter with reference to its tendency to communicate heat to matter in contact with it. If no heat flows upon contact, there is no difference in temperature.

Temperature, Absolute: Temperature expressed in degrees above absolute zero.

Temperature, Dry-Bulb: The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

Temperature, Dew-Point: The temperature at which the condensation of water vapor in a space begins for a given state of humidity and pressure as the temperature of the vapor is reduced. The temperature corresponding to saturation (100 per cent relative humidity) for a given absolute humidity at constant pressure.

Temperature, Effective: An arbitrary index which combines into a single value the effect of temperature, humidity, and air movement on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation.

Temperature, Wet-Bulb: Thermodynamic wet-bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet-bulb temperature (without qualification) is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications. (*A.S.M.E.* Power Test Codes, Series 1932, Instruments and Apparatus, Part 18.)

Thermodynamics, Laws of: Two laws upon which rest the classical theory of thermodynamics. These laws have been stated in many different, but equivalent ways. *The First Law*: (1) When work is expended in generating heat, the quantity of heat produced is proportional to the work expended; and conversely, when heat is employed in the performance of work, the quantity of heat which disappears is proportional to the work done. (Joule)[a] (G.P.)[b]; (2) If a system is caused to change from an initial state to a final state by adiabatic means only, the work done is the same for all adiabatic paths connecting the two states. (Zemansky); (3) In any power cycle or refrigeration cycle the net heat absorbed by the working substance is exactly equal to the net work done. *The Second Law*: (1) It is impossible for a self acting machine, unaided by any external agency, to convey heat from a body of lower to one of higher temperature. (Clausius) (G.P.); (2) It is impossible to derive mechanical work from heat taken from a body unless there is available a body of lower temperature into which the residue not so used may be discharged (Kelvin) (G.P.); (3) It is impossible to construct an engine that, operating in a cycle, will produce no effect other than the extraction of heat from a reservoir and the performance of an equivalent amount of work (Zemansky).

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls temperature.

Transmittance, Thermal: The time rate of heat flow, from the fluid on the warm side to the fluid on the cold side, per (square foot) (degree temperature difference between the two fluids). Sometimes called *Over-all Coefficient of Heat Transfer*.

Common unit is Btu per (hour) (square foot) (Fahrenheit degree). Symbol *U*.

Two-Pipe System (*Steam or Water*)**:** A heating system in which one pipe is used for the supply of the heating medium to the heating unit and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium which medium cannot have served a preceding heating unit.

Unit: As applied to heating, ventilating and air conditioning equipment this word means factory-built and assembled equipment with apparatus for accomplishing some specified function or combination of functions. (See Chapters 26 and 36.)

It is loosely applied to a great variety of equipment. Usually the function is included in the name, and hence come terms like Unit Heater, Unit Ventilator, Humidifying Unit, and Air Conditioning Unit.

Units are said to be *direct* or *room*, when intended for location, or located in, the treated space; *indirect* or *remote*, when outside or adjacent to the treated space. They are *ceiling* units when suspended from above, and *floor* when supported from below. Other descriptive words include *free delivery* when the unit is not intended to be attached to ducts or similar resistance-producing devices, and *pressure* when for use with such ducts. Complete description requires the use of several of these qualifying words or phases. (See Chapters 26 and 36.)

Up-Feed System: A heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vane Ratio: In air distributing devices the ratio of depth of vane to shortest opening width between two adjacent grille bars.

Vapor: The gaseous form of substances which are normally in the solid or liquid state and which can be changed to these states either by increasing the pressure or decreasing the temperature. Vapors diffuse. (*A.S.A.* definition.)

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[a]Names of authors who first stated law are given in parentheses.

[b]From Glossary of Physics, by LeRoy Dougherty Weld, (McGraw-Hill, 1937).

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensate to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry return.

Velocity: A vector quantity which denotes at once the time rate and the direction of a linear motion. $V = \frac{ds}{dt}$. For uniform linear motion $V = \frac{s}{t}$. Common units are: feet per second.

Ventilation: The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See Air Conditioning.)

Volume, Specific: The volume of a substance per unit mass; the reciprocal of density. Units: cubic feet per pound, cubic centimeters per gram, etc.

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts.

Warm Air Heating System, Gravity: A warm air heating system in which the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing.

Warm Air Heating System, Mechanical: A warm air heating system in which circulation of air is effected by a fan. Such a system may include air cleaning devices.

CHAPTER 2

Abbreviations and Symbols

Standard Abbreviations; Standard Symbols; Greek Alphabet; Conversion Equations; Graphical Symbols for Piping, Ductwork, Heating and Ventilating, Refrigerating; Identification of Piping by Color; Specific Heat Table

THIS chapter contains information regarding abbreviations, symbols, and conversion equations, which are of particular interest to the engineer engaged in heating, ventilating, and air conditioning.

ABBREVIATIONS

Abbreviations are shortened forms of names and expressions employed in texts and tabulations and should not generally be used as symbols in equations. Most of the following abbreviations have been compiled from a list of approved standards [1]. In general the period has been omitted in all abbreviations except where the omission results in the formation of an English word. Additional abbreviations applying to individual chapters will be found at the end of Chapters 3, 4, 5, 7, 16, 39, 41, and 43.

Absolute	abs
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
British thermal units per hour	Btuh
Calorie	cal
Centigram	cg
Centimeter	cm
Centimeter-gram-second (system)	cgs
Cubic	cu
Cubic centimeter	cu cm or cc
Cubic foot	cu ft
Cubic feet per minute	cfm
Cubic feet per second	cfs
Decibel	db
Degree[2]	deg or °
Degree, Centigrade	C
Degree, Fahrenheit	F
Degree, Kelvin	K
Degree, Réaumur	R
Diameter	diam

[1]Abbreviations for Scientific and Engineering Terms, Z10.1-1941 (*American Standards Association*).

[2]It is recommended that the abbreviation for the temperature scale, F, C, K, R, be included in expressions for numerical temperatures but, wherever feasible, the abbreviation for *degree* be omitted; as 68 F.

Direct-current (as adjective)	d-c
Electromotive Force	emf
Feet per minute	fpm
Feet per second	fps
Foot	ft
Foot-pound	ft-lb
Foot-pound-second (system)	fps
Freezing point	fp
Gallon	gal
Gallons per minute	gpm
Gallons per second	gps
Gram	g
Gram-calorie	g-cal
Horsepower	hp
Horsepower-hour	hp-hr
Hour	hr
Inch	in.
Inch-pound	in.-lb
Indicated horsepower	ihp
Indicated horsepower-hour	ihp-hr
Kilogram	kg
Kilowatt	kw
Kilowatthour	kwhr
Mass	mass
Melting point	mp
Meter	m
Micron	μ (mu)
Miles per hour	mph
Millimeter	mm
Minute	min
Molecular weight	mol. wt
Mol	mol
Ounce	oz
Pound	lb
Pounds per square inch	psi
Pounds per square inch, gage	psig
Pounds per square inch, absolute	psia
Revolutions per minute	rpm
Revolutions per second	rps
Second	sec
Specific gravity	sp gr
Specific heat	sp ht
Square foot	sq ft
Square inch	sq in.
Watt	w
Watthour	whr

SYMBOLS

A letter symbol is a single character, with subscript or superscript if required, used to designate a physical magnitude in mathematical equations and expressions. Two or more symbols together always represent a product. The following have been compiled from a selected list of approved standards [3]. Additional symbols and variations in the standard

[3]Letter Symbols for Mechanics of Solid Bodies, Z10.3-1942, and Letter Symbols for Heat and Thermodynamics, Z10.4-1943 (*American Standards Association*).

symbols found necessary in the individual chapters will be found in a list at the end of Chapters 3, 4, 5, 7, 16, 39, 41, and 43.

Acceleration, due to gravity........ g
Acceleration, linear........ a
Area........ A
Change in specific volume during vaporization........ v_{fg}
Density, Weight per unit volume, Specific weight........ d or ρ (rho)

$$d = \frac{1}{v}$$

Distance, linear........ s
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, vapor in contact with liquid........ *Subscript* g

Efficiency........ η
Elevation above some datum........ z, Z
Emissivity........ ε
Energy in general; work, total; work, molal........ E
Entropy. (The capital should be used for any weight, and the small letter for unit weight)........ S or s

Force, total load........ F
Gas Constant, in equation $pV = nRT$........ R
Head........ H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight)........ H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid........ h_f

Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor........ h_g
Heat of vaporization at constant pressure........ L or h_{fg}
Hydraulic radius........ R_H
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight)........ U or u
Length of path of heat flow, thickness........ L

Load, total........ W
Mechanical efficiency........ e_m
Mechanical equivalent of heat........ J
Power, Horsepower, Work per unit time........ P
Pressure, Absolute pressure, Gage pressure, Force per unit area........ p

Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred........ Q
Quality of steam, Pounds of dry steam per pound of mixture........ x
Reynolds Number........ N_{Re}
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor........ *Subscript* f
Specific heat........ c

Specific heat at constant pressure........ c_p
Specific heat at constant volume........ c_v
Specific volume, Volume per unit weight, Volume per unit mass........ v
Temperature (ordinary) F or C. (*Theta* is used preferably only when t is used for Time in the same discussion)........ t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature)........ T or Θ (*capital theta*)

Thermal conductance[4]: heat transferred per (unit time) (degree) C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

Thermal conductance per unit area, Unit conductance: heat transferred per (unit time) (unit area) (degree) .... C_a

$$C_a = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

Thermal conductivity: heat transferred per (unit time) (unit area) (degree per unit length) .... k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer: heat transferred per (unit time) (unit area) (degree) .... f

$$f = \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)

Over-all coefficient of heat transfer, Thermal transmittance per unit area: heat transferred per (unit time) (unit area) (degree over-all) .... U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

Thermal transmission (heat transferred per unit time) .... q

$$q = \frac{Q}{t}$$

Thermal resistance (degree per unit of heat transferred per unit time) .... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity .... $1/k$

Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure .... *Subscript* fg

Velocity V

Viscosity, absolute .... μ

Viscosity, kinematic .... μ/ρ

Volume (total) V

Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight .... q

Weight of a major item, Total weight .... W

Weight rate, Weight per unit of power, Weight per unit of time .... w

Work (total) .... W

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[4]Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties*. Examples: conductivity, resistivity. Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities*. Examples conductance, transmittance. Terms ending *ion* designate rate of heat transfer. Examples conduction, transmission

THE GREEK ALPHABET

Α α	Alpha	Ι ι	Iota	Ρ ρ	Rho
Β β	Beta	Κ κ	Kappa	Σ σ s	Sigma
Γ γ	Gamma	Λ λ	Lambda	Τ τ	Tau
Δ δ	Delta	Μ μ	Mu	Υ υ	Upsilon
Ε ε	Epsilon	Ν ν	Nu	Φ φ	Phi
Ζ ζ	Zeta	Ξ ξ	Xi	Χ χ	Chi
Η η	Eta	Ο ο	Omicron	Ψ ψ	Psi
Θ ϑ θ	Theta	Π π	Pi	Ω ω	Omega

CONVERSION EQUATIONS[5]

Heat, Power and Work

1 ton refrigeration	= 12,000 Btu per hour 200 Btu per minute
Latent heat of ice	= 143.4 Btu per pound
1 Btu	= 778.3 ft-lb 0.2930 Int. whr 252.0 I.T. calorie
1 Int. watthour	= 2656 ft-lb 3.413 Btu 3600 Int. joules 860 I.T. calories
1 Int. kilowatthour	= 3,413 Btu 3.517 lb water evaporated from and at 212 F
1 Int. kilowatt (1000 watts)	= 1 341 hp 56.88 Btu per minute 44,267 ft-lb per minute
1000 I.T. calories 1 I.T. Kilocalorie	= 3.968 Btu 3088 ft-lb 1.1628 Int. whr
1 horsepower	= 0.7455 Int. kw 42.40 Btu per minute 33,000 ft-lb per minute 550 ft-lb per second
1 boiler horsepower	= 33,475 Btu per hour 9.809 Int. kw

Weight and Volume

1 gal (U. S.)	= 231 cu in. 0.1337 cu ft
1 British or Imperial gallon	= 277.42 cu in.
1 cu ft	= 7.481 gal 1728 cu in.
1 cu ft water at 60 F (in vacuo)	= 62.37 lb
1 cu ft water at 212 F (" ")	= 59.83 lb
1 gal water at 60 F (" ")	= 8.338 lb
1 gal water at 212 F (" ")	= 7.998 lb
1 lb (avdp)	= 16 oz 7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb

[5]Checked in 1944 by *National Bureau of Standards*. Abbreviations *Int.* and *I.T.* refer to *International* and *International (Steam) Table* respectively.

Pressure

1 lb per square inch	= 144 lb per square foot 2.0360 in mercury at 32 F 2.0422 in. mercury at 62 F 2.309 ft water at 62 F 27.71 in. water at 62 F
1 oz per square inch	= 0.1276 in. mercury at 62 F 1.732 in. water at 62 F
1 atmosphere	= 14.696 lb per square inch 2116 lb per square foot 33.94 ft water at 62 F 30.01 in. mercury at 62 F 29.921 in. mercury at 32 F
1 in. water at 62 F (in vacuo)	= 0.03609 lb per square inch 0.5774 oz per square inch 5.197 lb per square foot
1 ft water at 62 F (in vacuo)	= 0.4330 lb per square inch 62.37 lb per square foot
1 in. mercury at 62 F (in vacuo)	= 0.4897 lb per square inch 7.835 oz per square inch 1.131 ft water at 62 F 13.57 in. water at 62 F
1 in. mercury at 32 F (in vacuo)	= 0.49115 lb per square inch

Metric Units

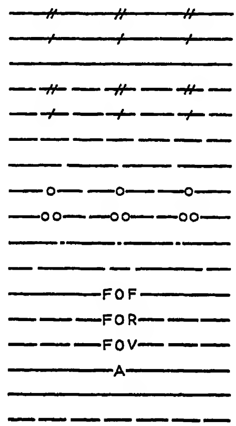
1 cm	= 0.3937 in. = 0.0328 ft
1 in.	= 2.540 cm
1 m	= 3.281 ft
1 ft	= 0.3048 m
1 sq cm	= 0.1550 sq in.
1 sq in.	= 6.452 sq cm
1 sq m	= 10.76 sq ft
1 sq ft	= 0.09290 sq m
1 cu cm	= 0.06102 cu in.
1 cu in.	= 16.39 cu cm
1 cu m	= 35.31 cu ft
1 cu ft	= 0.02832 cu m
1 liter	= 1000 cu cm = 0.2642 gal
1 kg	= 2.205 lb (avdp)
1 lb	= 0.4536 kg
1 metric ton	= 2205 lb (avdp)
1 gram	= 0.002205 lb (avdp)
1 kilometer per hour	= 0.6214 mph
1 gram per square centimeter	= 0.02905 in. mercury at 62 F 0.3942 in. water at 62 F
1 kg per sq cm (metric atmosphere)	= 14.22 lb per square inch
1 gram per cubic centimeter	= 0.03613 lb per cubic inch 62.43 lb per cubic foot
1 dyne	= 0.00007233 poundals
1 absolute joule	= 10,000,000 ergs 0.7376 ft-lb
1 Int. joule	= 0.7378 ft-lb
1 metric horsepower	= 75 kg-m per second 0.986 hp (U. S.)
1 I. T. kilocalorie per kilogram	= 1.8 Btu per pound
1 I.T. calorie per square centimeter	= 3.687 Btu per square foot
1 I.T. calorie per (second) (square centimeter) for a temperature gradient of 1 C deg per centimeter	= 2903 Btu per (hour) (square foot) for a temperature gradient of 1 F deg per inch of thickness.

GRAPHICAL SYMBOLS FOR DRAWINGS[6]

Graphical Symbols for Drawings *Piping*

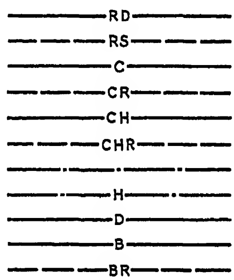
Heating

1. High Pressure Steam
2. Medium Pressure Steam
3. Low Pressure Steam
4. High Pressure Return
5. Medium Pressure Return
6. Low Pressure Return
7. Boiler Blow Off
8. Condensate or Vacuum Pump Discharge
9. Feedwater Pump Discharge
10. Make Up Water
11. Air Relief Line
12. Fuel Oil Flow
13. Fuel Oil Return
14. Fuel Oil Tank Vent
15. Compressed Air
16. Hot Water Heating Supply
17. Hot Water Heating Return



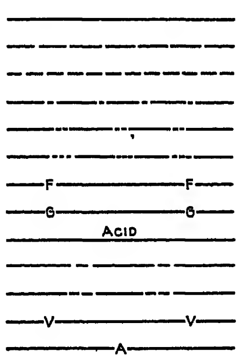
Air Conditioning

18. Refrigerant Discharge
19. Refrigerant Suction
20. Condenser Water Flow
21. Condenser Water Return
22. Circulating Chilled or Hot Water Flow
23. Circulating Chilled or Hot Water Return
24. Make Up Water
25. Humidification Line
26. Drain
27. Brine Supply
28. Brine Return



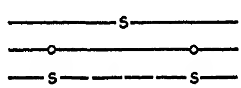
Plumbing

29. Soil, Waste or Leader (Above Grade)
30. Soil, Waste or Leader (Below Grade)
31. Vent
32. Cold Water
33. Hot Water
34. Hot Water Return
35. Fire Line
36. Gas
37. Acid Waste
38. Drinking Water Flow
39. Drinking Water Return
40. Vacuum Cleaning
41. Compressed Air



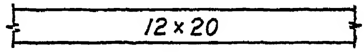
Sprinklers

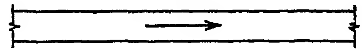
42. Main Supplies
43. Branch and Head
44. Drain

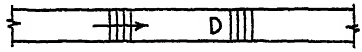


[6] Graphical Symbols for Use on Drawings in Mechanical Engineering, Z32.2-1941 (*American Standards Association*).

Graphical Symbols for Drawings *Ductwork*

45. Duct (1st Figure, Width; 2nd, Depth)


46. Direction of Flow


47. Inclined Drop in Respect to Air Flow


48. Inclined Rise in Respect to Air Flow


49. Supply Duct Section


50. Exhaust Duct Section


51. Recirculation Duct Section

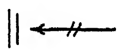

52. Fresh Air Duct Section

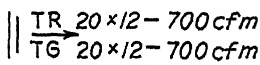

53. Other Duct Sections

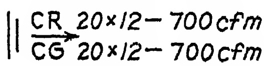

54. Register — R

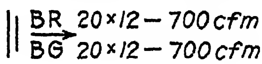
55. Grille — G

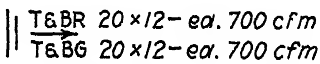
56. Supply Outlet

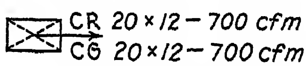

57. Exhaust Inlet


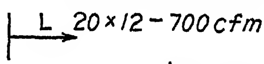
58. Top Register or Grille


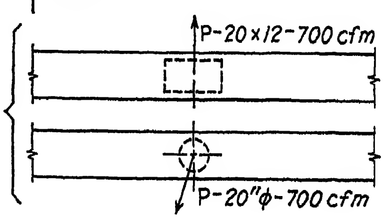
59. Center Register or Grille


60. Bottom Register or Grille


61. Top and Bottom Register or Grille


62. Ceiling Register or Grille


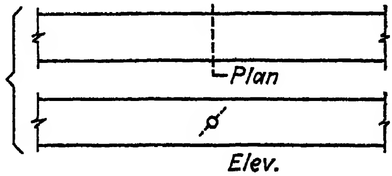
63. Louver Opening


64. Adjustable Plaque


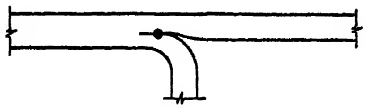
GRAPHICAL SYMBOLS FOR DRAWINGS

Ductwork

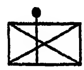
65. Volume Damper



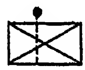
66. Deflecting Damper



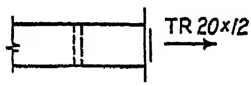
67. Deflecting Damper, Up



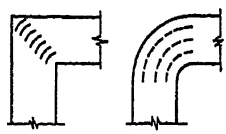
68. Deflecting Damper, Down



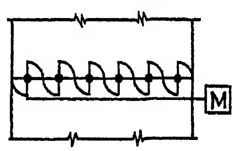
69. Adjustable Blank Off



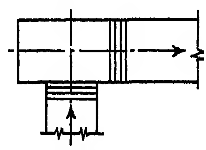
70. Turning Vanes



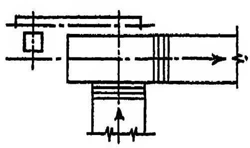
71. Automatic Dampers



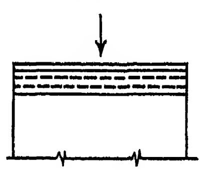
72. Canvas Connections



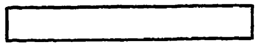
73. Fan and Motor With Guard



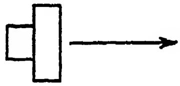
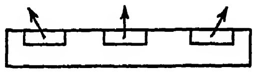
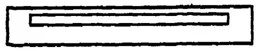
74. Intake Louvers and Screen



Graphical Symbols for Drawings *Heating and Ventilating*

75. Heat Transfer Surface, Plan

76. Wall Radiator, Plan

77. Wall Radiator on Ceiling, Plan

78. Unit Heater (Propeller), Plan

79. Unit Heater (Centrifugal Fan), Plan

80. Unit Ventilator, Plan


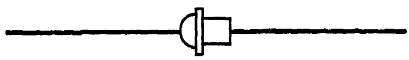
Traps

81. Thermostatic

82. Blast Thermostatic

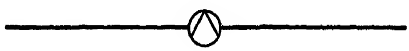
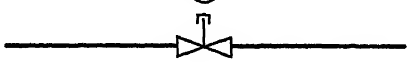
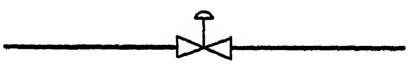
83. Float and Thermostatic

84. Float

85. Boiler Return


Valves

86. Reducing Pressure

87. Air Line

88. Lock and Shield

89. Diaphragm

90. Air Eliminator

91. Strainer

92. Thermometer

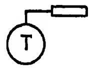
93. Thermostat


Graphical Symbols for Drawings

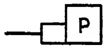
Refrigerating

94. Thermostat (Self Contained)

95. Thermostat (Remote Bulb)



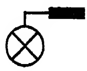
96. Pressurestat



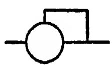
97. Hand Expansion Valve

98. Automatic Expansion Valve

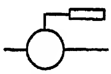
99. Thermostatic Expansion Valve



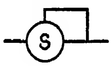
100. Evaporator Press. Regulating Valve, Throttling Type



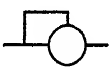
101. Evaporator Press. Regulating Valve, Thermostatic Throttling Type



102. Evaporator Press. Regulating Valve, Snap-Action Valve



103. Compressor Suction Pressure Limiting Valve, Throttling Type



104. Hand Shut Off Valve

105. Thermal Bulb

106. Scale Trap

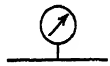
107. Dryer

108. Strainer

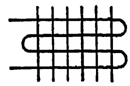
109. High Side Float

110. Low Side Float

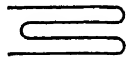
111. Gage



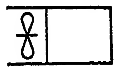
112. Finned Type Cooling Unit, Natural Convection



113. Pipe Coil



114. Forced Convection Cooling Unit



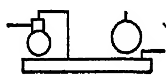
115. Immersion Cooling Unit

116. Ice Making Unit

117. Heat Interchanger

118. Condensing Unit, Air Cooled

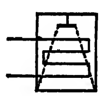
119. Condensing Unit, Water Cooled



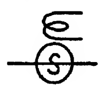
120. Compressor

121. Cooling Tower

122. Evaporative Condenser



123. Solenoid Valve



124. Pressurestat With High Pressure Cut-Out

IDENTIFICATION OF PIPING SYSTEMS BY COLOR

The color scheme for identification of piping systems listed in the following table and shown in Fig. 1 is reprinted from Part V, Fourth Edition, of the Engineering Standards of the *Heating, Piping and Air Conditioning Contractors National Association*[7].

All piping systems are classified according to the material carried in the pipes and colors are assigned as follows:

CLASS	COLOR
F—Fire-protection	Red
D—Dangerous materials	Yellow or Orange
S—Safe Materials	Green (or the achromatic colors, white, black, gray or aluminum)
and, when required	
P—Protective materials	Bright blue
V—Extra valuable materials	Deep purple

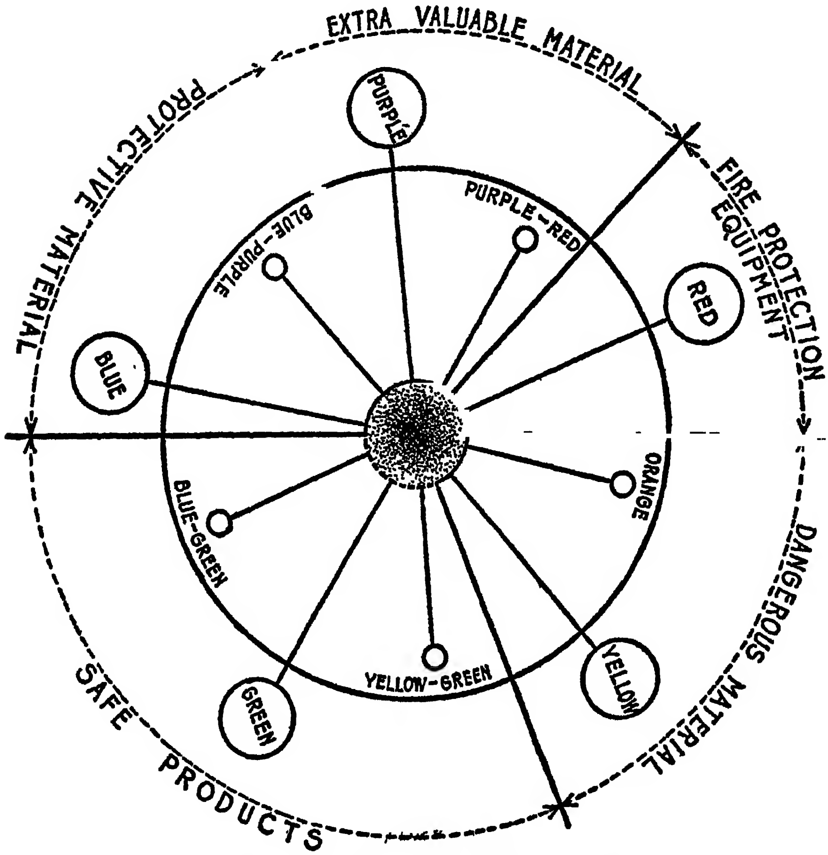


FIG. 1. MAIN CLASSIFICATION BY COLOR[a]

[a]From Scheme for Identification of Piping Systems, *Heating, Piping and Air Conditioning Contractors National Association*, Part V, Fourth Edition, p. 17. Used by permission.

[7]See Scheme for Identification of Piping Systems, A13-1928, *American Standards Association*.

SPECIFIC HEAT

Table 1. Specific Heat of Solids[a]

Materials	Temperature F	Specific Heat	Authority
Alloys			
Brass, Red	32	0.0899	S
Brass, Yellow	32	0.0883	S
Bronze (80*Cu*, 20*Sn*)	57–208	0.0862	S
Monel Metal	68–2370	0.127	S
Aluminum	80–212	0.212	S
Asbestos	68–208	0.195	S
Brickwork	——	0.195	H
Carbon (Graphite)	104–1637	0.314	I
Coal	——	0.278	H
Coke	——	0.201	H
Concrete	——	0.270	H
Copper	64–212	0 0928	S
Fire Clay Brick	77–1832	0 258	I
Glass			
Crown	50–122	0.161	S
Flint	50–122	0.117	S
Gold	64	0.0312	S
Gypsum	——	0.259	H
Ice	32	0.487	S
Ice	−40	0.434	S
Iron, Pure	32	0 1043	S
Iron, Pure	32–600	0 127	M
Iron, Cast	68–212	0.1189	H
Iron, Wrought	59–212	0.1152	H
Lead	32	0 0297	S
Nickel	32	0.1032	S
Masonry	——	0.2159	H
Plaster	——	0.2	H
Platinum	58–212	0.0319	S
Rocks			
Gneiss	63–210	0 196	S
Granite	54–212	0.192	S
Limestone	59–212	0.216	S
Marble	32–212	0.21	S
Sandstone	——	0 22	S
Silver	32	0 0536	S
Steel	——	0.1175	H
Sulphur	240–320	0.220	S
Silica Brick	77–1832	0.263	I
Tin	77	0.0548	S
Woods (Average)	68	0 327	S
Zinc	32	0.0913	S

Table 2. Specific Heat of Liquids

Liquid	Temperature F	Specific Heat	Authority
Alcohol, Ethyl	32	0.548	S
Alcohol, Methyl	59–68	0.601	S
Glycerine	59–122	0.576	S
Lead (Molten)	360	0 041	H
Mercury	68	0 03325	S
Petroleum	70–136	0.511	S
Sea Water			
Sp Gr 1.0043	64	0 980	S
Sp Gr 1 0463	64	0.903	S
Water	59	1.000	S

Table 3. Specific Heat of Gases and Vapors

Substance	Temperature F	Specific Heat at Constant Pressure	Ratio of Specific Heat C_p/C_v	Specific Heat at Constant Volume (Computed)	Authority
Air	32–392	0 2375	1.405	0.169	S
Ammonia	80–392	0.5356	1.277	0.419	S
Carbon Dioxide	52–417	0.2169	1.3003	0.1668	S
Carbon Monoxide	79–388	0.2426	1.395	0.1736	S
Coal Gas	68–1900	0.3145	——	——	S
Flue Gas	——	0 24 (Approx.)	——	——	H
Hydrogen	70–212	3.41	1.419	2.402	S
Nitrogen	32–392	0.2438	1.41	0.1729	S
Oxygen	55–404	0.2175	1.3977	0.155	S
Water Vapor	212	0.421	1.305	0.322	S
Water Vapor	356	0.51	——	——	S

[a]See also The Specific Heat of Thermal Insulating Materials, by Gordon B. Wilkes and Carl O. Wood (A.S.H.V.E. Transactions, Vol. 48, 1942, p. 493).

Notes: When one temperature is given the true specific heat is given, otherwise the value is the mean specific heat between the given limits.

Authorities: S—Smithsonian Physical Tables, 1933; I—International Critical Tables; H—Heating, Ventilation and Air Conditioning, by L. A. Harding and A. C. Willard; M—Engineers' Handbook, by Lionel S. Marks

CHAPTER 3

Thermodynamics

Degree of Saturation; Mollier Diagram; Derived Properties; Typical Air Conditioning Processes, Heating, Cooling, Adiabatic Mixing; Wet-Bulb Temperatures Below 32 F; Dalton's Rule; Steady Flow Energy Equation, U. S. Standard Atmosphere

THE working substance of the air conditioning engineer is called *moist air*. In order to be able to apply the laws of conservation of energy and mass to the analysis of typical air conditioning processes, it is necessary to know the thermodynamic properties of moist air, particularly its enthalpy and volume. When the limitations imposed by the Second Law of Thermodynamics have to be considered, it is also necessary to know its entropy.

For the purpose of analysis, moist air may be regarded as a mixture of only two constituents, namely, dry air and water vapor. It has long been customary to predict the thermodynamic properties of the mixture from a knowledge of those of dry air and water vapor separately by means of Dalton's Rule. According to this rule: each constituent of a gas mixture occupies the whole volume of the mixture just as if no other constituent were present; it therefore exerts a *partial pressure* equal to the pressure it would exert if alone in the whole volume at the temperature of the mixture; the observed pressure of the mixture is the sum of these so-called partial pressures; the enthalpy of the mixture is the sum of separate contributions from the individual constituents as determined by their partial pressures, their weights, and the temperature of the mixture; and the entropy of the mixture is obtained in a similar manner.

Dalton's Rule has long been regarded erroneously as a fundamental law of nature. Actually it is not, and in many cases its predictions are quite unreliable. In the case of moist air at atmospheric pressure, it happens to give a close approximation to the truth; but as progress is made the need for greater accuracy than the rule can afford is felt even in this case. Fortunately most of the complications involved in following a correct procedure based on the predictions of statistical mechanics are met in preparing suitable tables of thermodynamic properties; and, once these tables have been prepared, their use in the analysis of typical air conditioning processes is actually simplified by abandonment of the rule together with its fictitious concepts of partial pressure, relative humidity, etc.

Thermodynamic Properties of Moist Air

Table 1, Thermodynamic Properties of Moist Air (Standard Atmospheric Pressure, 29.921 In. Hg), contains results of a cooperative investigation between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Towne Scientific School, University of Pennsylvania. These results are to be considered by an International Joint Committee on Psychrometric Data as a possible starting point from which to reach agreement on standard properties of moist air. A detailed explanation of the data and methods used in constructing Table 1 is given in a paper [1] presented before the ASHVE upon the recommendation of its Technical

Advisory Committee on Psychrometry as a final report of the cooperative investigation.

In Table 1 there are 15 columns of figures, each column being headed by a suitable symbol. In the following sub-paragraphs are given brief explanations of the data in Table 1 under the appropriate column headings.

t(F) = Fahrenheit temperature defined in terms of absolute temperature T by the relation,

$$T = t + 459.69 \tag{1}$$

This particular Fahrenheit scale differs slightly from that derived from the International Centigrade Scale t(C) by the definition,

$$t(\mathrm{F}) = 1.8t(\mathrm{C}) + 32 \tag{2}$$

However, the maximum difference between the two Fahrenheit scales appears not to exceed 0.01 Fahrenheit deg in the range 32 to 212 F.

W_s = humidity ratio *at saturation*. By humidity ratio is meant the ratio, by weight, of water vapor to dry air, pounds of water vapor per pound of dry air. By saturation is meant the point where coexistence of the vapor phase (moist air) with a condensed phase (liquid or solid) is possible at the given temperature and pressure (standard atmospheric pressure in the case of Table 1). At given values of temperature and pressure the humidity ratio W can have any value from zero to W_s.

v_a = specific volume of dry air, cubic feet per pound.

$v_{as} = v_s - v_a$, the difference between the volume of moist air *at saturation*, per pound of dry air, and the specific volume of the dry air itself, cubic feet per pound of dry air.

v_s = volume of moist air *at saturation* per pound of dry air, cubic feet per pound of dry air.

h_a = specific enthalpy of dry air, Btu per pound. It will be noticed that the specific enthalpy of dry air has been assigned the value zero at 0 F, standard atmospheric pressure. The energy unit Btu is related to the foot-pound, though not exactly, by definition, as follows: 1 Btu = 778.18 ft-lb.

$h_{as} = h_s - h_a$, the difference between the enthalpy of moist air *at saturation*, per pound of dry air, and the specific enthalpy of the dry air itself, Btu per pound of dry air.

h_s = enthalpy of moist air *at saturation* per pound of dry air, Btu per pound of dry air.

s_a = specific entropy of dry air, Btu per (pound) (F). It will be noticed that the specific entropy of dry air has been assigned the value zero at 0 F and standard atmospheric pressure.

$s_{as} = s_s - s_a$, the difference between the entropy of moist air *at saturation*, per pound of dry air, and the specific entropy of the dry air itself, Btu per (pound of dry air) (F).

s_s = entropy of moist air *at saturation* per pound of dry air, Btu per (pound of dry air) (F).

h_w = specific enthalpy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per pound of water. The specific enthalpy of liquid water has been assigned the value zero at 32 F, saturation pressure. It will be noticed that, under this assignment, the specific enthalpy of liquid water at 32 F, standard atmospheric pressure, assumes the value 0.04 Btu/$\mathrm{lb_w}$.

s_w = specific entropy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per (pound of water) (F). The specific entropy of liquid water has been assigned the value zero at 32 F, saturation pressure. It will be noticed that, under this assignment, the specific entropy of liquid water at 32 F, standard atmospheric pressure, is also zero, though not exactly.

p_s = saturation pressure of pure water vapor, pounds per square inch or inches of Hg. Moist air can be saturated at any given values of temperature and pressure, though this requires that it have a definite humidity ratio W_s and that the coexisting condensed phase contain a definite, but very small quantity of dissolved air. On the other hand, pure water vapor (steam) cannot be saturated at any given values of temperature and pressure because its composition is invariable. It can, however, be saturated at any

Table 1. Thermodynamic Properties of Moist Air[a] (Standard Atmospheric Pressure, 29.921 in. Hg)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^3$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap Press In. Hg $p_s \times 10^6$	
−160	0.2120	7.520	0.000	7.520	−38.504	0.000	−38.504	−0.10300	0.00000	−0.10300	−222.00	−0.4907	0.1009	−160
−159	0.2394	7.545	0.000	7.545	−38.262	0.000	−38.262	−0.10219	0.00000	−0.10219	−221.68	−0.4896	0.1139	−159
−158	0.2703	7.571	0.000	7.571	−38.021	0.000	−38.021	−0.10139	0.00000	−0.10139	−221.36	−0.4885	0.1286	−158
−157	0.3049	7.596	0.000	7.596	−37.779	0.000	−37.779	−0.10059	0.00000	−0.10059	−221.04	−0.4874	0.1450	−157
−156	0.3435	7.622	0.000	7.622	−37.538	0.000	−37.538	−0.09980	0.00000	−0.09980	−220.72	−0.4864	0.1635	−156
−155	0.3869	7.647	0.000	7.647	−37.296	0.000	−37.296	−0.09901	0.00000	−0.09901	−220.40	−0.4853	0.1842	−155
−154	0.4354	7.673	0.000	7.673	−37.055	0.000	−37.055	−0.09822	0.00000	−0.09822	−220.07	−0.4843	0.2073	−154
−153	0.4897	7.699	0.000	7.699	−36.813	0.000	−36.813	−0.09743	0.00000	−0.09743	−219.75	−0.4832	0.2331	−153
−152	0.5502	7.724	0.000	7.724	−36.572	0.000	−36.572	−0.09664	0.00000	−0.09664	−219.42	−0.4822	0.2620	−152
−151	0.6178	7.750	0.000	7.750	−36.330	0.000	−36.330	−0.09586	0.00000	−0.09586	−219.10	−0.4811	0.2942	−151
−150	0.6932	7.775	0.000	7.775	−36.088	0.000	−36.088	−0.09508	0.00000	−0.09508	−218.77	−0.4800	0.3301	−150
−149	0.7772	7.801	0.000	7.801	−35.847	0.000	−35.847	−0.09430	0.00000	−0.09430	−218.44	−0.4789	0.3701	−149
−148	0.8709	7.826	0.000	7.826	−35.606	0.000	−35.606	−0.09352	0.00000	−0.09352	−218.11	−0.4779	0.4146	−148
−147	0.9750	7.851	0.000	7.851	−35.364	0.000	−35.364	−0.09274	0.00000	−0.09274	−217.78	−0.4768	0.4641	−147
−146	1.091	7.876	0.000	7.876	−35.123	0.000	−35.123	−0.09198	0.00000	−0.09198	−217.45	−0.4758	0.5194	−146
−145	1.219	7.902	0.000	7.902	−34.881	0.000	−34.881	−0.09121	0.00000	−0.09121	−217.12	−0.4747	0.5807	−145
−144	1.362	7.927	0.000	7.927	−34.640	0.000	−34.640	−0.09044	0.00000	−0.09044	−216.78	−0.4737	0.6488	−144
−143	1.521	7.953	0.000	7.953	−34.399	0.000	−34.399	−0.08967	0.00000	−0.08967	−216.45	−0.4726	0.7243	−143
−142	1.698	7.978	0.000	7.978	−34.157	0.000	−34.157	−0.08892	0.00000	−0.08892	−216.11	−0.4716	0.8082	−142
−141	1.893	8.004	0.000	8.004	−33.916	0.000	−33.916	−0.08816	0.00000	−0.08816	−215.78	−0.4705	0.9011	−141
−140	2.109	8.029	0.000	8.029	−33.674	0.000	−33.674	−0.08740	0.00000	−0.08740	−215.44	−0.4695	1.004	−140
−139	2.348	8.054	0.000	8.054	−33.433	0.000	−33.433	−0.08664	0.00000	−0.08664	−215.11	−0.4684	1.118	−139
−138	2.613	8.079	0.000	8.079	−33.192	0.000	−33.192	−0.08589	0.00000	−0.08589	−214.77	−0.4674	1.244	−138
−137	2.906	8.105	0.000	8.105	−32.951	0.000	−32.951	−0.08514	0.00000	−0.08514	−214.43	−0.4663	1.383	−137
−136	3.229	8.130	0.000	8.130	−32.709	0.000	−32.709	−0.08440	0.00000	−0.08440	−214.09	−0.4653	1.537	−136
−135	3.586	8.156	0.000	8.156	−32.468	0.000	−32.468	−0.08365	0.00000	−0.08365	−213.75	−0.4642	1.707	−135
−134	3.980	8.181	0.000	8.181	−32.226	0.000	−32.226	−0.08291	0.00000	−0.08291	−213.41	−0.4632	1.895	−134
−133	4.414	8.207	0.000	8.207	−31.985	0.000	−31.985	−0.08217	0.00000	−0.08217	−213.07	−0.4621	2.102	−133
−132	4.893	8.232	0.000	8.232	−31.744	0.000	−31.744	−0.08144	0.00000	−0.08144	−212.72	−0.4611	2.330	−132
−131	5.419	8.258	0.000	8.258	−31.503	0.000	−31.503	−0.08070	0.00000	−0.08070	−212.38	−0.4600	2.581	−131
−130	6.000	8.283	0.000	8.283	−31.262	0.000	−31.262	−0.07997	0.00000	−0.07997	−212.03	−0.4590	2.858	−130
−129	6.637	8.309	0.000	8.309	−31.021	0.000	−31.021	−0.07924	0.00000	−0.07924	−211.68	−0.4579	3.182	−129

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^7$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water: Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In Hg $p_s \times 10^5$	Fahr. Temp. t(F)
−128	0.7339	8.334	0.000	8.334	−30.780	0.000	−30.780	−0.07851	0.00000	−0.07851	−211.33	−0.4569	0.3492	−128
−127	0.8111	8.360	0.000	8.360	−30.539	0.000	−30.539	−0.07778	0.00000	−0.07778	−210.98	−0.4559	0.3863	−127
−126	0.8958	8.385	0.000	8.385	−30.298	0.000	−30.298	−0.07707	0.00000	−0.07707	−210.63	−0.4548	0.4267	−126
−125	0.9887	8.411	0.000	8.411	−30.057	0.000	−30.057	−0.07634	0.00000	−0.07634	−210.28	−0.4538	0.4710	−125
−124	1.091	8.436	0.000	8.436	−29.816	0.000	−29.816	−0.07562	0.00000	−0.07562	−209.93	−0.4527	0.5197	−124
−123	1.202	8.461	0.000	8.461	−29.575	0.000	−29.575	−0.07490	0.00000	−0.07490	−209.58	−0.4517	0.5730	−123
−122	1.325	8.486	0.000	8.486	−29.334	0.000	−29.334	−0.07419	0.00000	−0.07419	−209.23	−0.4506	0.6314	−122
−121	1.459	8.512	0.000	8.512	−29.093	0.000	−29.093	−0.07348	0.00000	−0.07348	−208.88	−0.4496	0.6953	−121
−120	1.606	8.537	0.000	8.537	−28.852	0.000	−28.852	−0.07277	0.00000	−0.07277	−208.52	−0.4485	0.7653	−120
−119	1.767	8.563	0.000	8.563	−28.611	0.000	−28.611	−0.07206	0.00000	−0.07206	−208.17	−0.4475	0.8419	−119
−118	1.942	8.588	0.000	8.588	−28.370	0.000	−28.370	−0.07135	0.00000	−0.07135	−207.81	−0.4464	0.9256	−118
−117	2.134	8.613	0.000	8.613	−28.129	0.000	−28.129	−0.07064	0.00000	−0.07064	−207.45	−0.4454	1.017	−117
−116	2.343	8.639	0.000	8.639	−27.889	0.000	−27.889	−0.06994	0.00000	−0.06994	−207.09	−0.4444	1.117	−116
−115	2.571	8.664	0.000	8.664	−27.648	0.000	−27.648	−0.06924	0.00000	−0.06924	−206.73	−0.4433	1.226	−115
−114	2.820	8.690	0.000	8.690	−27.407	0.000	−27.407	−0.06854	0.00000	−0.06854	−206.37	−0.4423	1.345	−114
−113	3.092	8.715	0.000	8.715	−27.166	0.000	−27.166	−0.06784	0.00000	−0.06784	−206.01	−0.4412	1.475	−113
−112	3.388	8.741	0.000	8.741	−26.926	0.000	−26.926	−0.06715	0.00000	−0.06715	−205.65	−0.4402	1.617	−112
−111	3.711	8.766	0.000	8.766	−26.685	0.000	−26.685	−0.06646	0.00000	−0.06646	−205.29	−0.4392	1.771	−111
−110	4.063	8.792	0.000	8.792	−26.444	0.000	−26.444	−0.06577	0.00000	−0.06577	−204.92	−0.4381	1.939	−110
−109	4.445	8.817	0.000	8.817	−26.204	0.000	−26.204	−0.06508	0.00000	−0.06508	−204.46	−0.4371	2.121	−109
−108	4.861	8.842	0.000	8.842	−25.963	0.001	−25.962	−0.06439	0.00000	−0.06439	−204.19	−0.4360	2.320	−108
−107	5.314	8.868	0.000	8.868	−25.772	0.001	−25.721	−0.06370	0.00000	−0.06370	−203.83	−0.4350	2.536	−107
−106	5.806	8.893	0.000	8.893	−25.481	0.001	−25.480	−0.06302	0.00000	−0.06302	−203.46	−0.4339	2.771	−106
−105	6.340	8.919	0.000	8.919	−25.240	0.001	−25.239	−0.06234	0.00000	−0.06234	−203.09	−0.4329	3.026	−105
−104	6.920	8.944	0.000	8.944	−25.000	0.001	−24.999	−0.06167	0.00000	−0.06167	−202.72	−0.4318	3.303	−104
−103	7.549	8.970	0.000	8.970	−24.759	0.001	−24.758	−0.06099	0.00000	−0.06099	−202.35	−0.4308	3.603	−103
−102	8.232	8.995	0.000	8.995	−24.518	0.001	−24.517	−0.06032	0.00000	−0.06032	−201.98	−0.4298	3.929	−102
−101	8.972	9.020	0.000	9.020	−24.278	0.001	−24.277	−0.05964	0.00000	−0.05964	−201.61	−0.4287	4.283	−101
−100	9.772	9.046	0.000	9.046	−24.037	0.001	−24.036	−0.05897	0.00000	−0.05897	−201.23	−0.4277	4.666	−100
−99	10.63	9.071	0.000	9.071	−23.797	0.001	−23.796	−0.05830	0.00000	−0.05830	−200.86	−0.4266	5.081	−99
−98	11.57	9.097	0.000	9.097	−23.556	0.001	−23.555	−0.05764	0.00000	−0.05764	−200.48	−0.4256	5.530	−98
−97	12.59	9.122	0.000	9.122	−23.316	0.001	−23.315	−0.05697	0.00000	−0.05697	−200.11	−0.4245	6.016	−97

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TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^6$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb) (°F) s_w	Vap. Press In. Hg $p_s \times 10^4$	
−96	1.369	9.147	0.000	9.147	−23.075	0.001	−23.074	−0.05631	0.00000	−0.05631	−199.73	−0.4235	0.6542	−96
−95	1.489	9.173	0.000	9.173	−22.835	0.002	−22.833	−0.05565	0.00000	−0.05565	−199.35	−0.4225	0.7111	−95
−94	1.617	9.198	0.000	9.198	−22.594	0.002	−22.592	−0.05500	0.00000	−0.05500	−198.97	−0.4214	0.7725	−94
−93	1.756	9.224	0.000	9.224	−22.353	0.002	−22.341	−0.05434	0.00000	−0.05434	−198.59	−0.4204	0.8388	−93
−92	1.906	9.249	0.000	9.249	−22.113	0.002	−22.111	−0.05369	0.00001	−0.05368	−198.21	−0.4193	0.9105	−92
−91	2.068	9.275	0.000	9.275	−21.872	0.002	−21.870	−0.05303	0.00001	−0.05302	−197.83	−0.4183	0.9879	−91
−90	2.242	9.300	0.000	9.300	−21.631	0.002	−21.629	−0.05237	0.00001	−0.05236	−197.44	−0.4173	1.071	−90
−89	2.430	9.325	0.000	9.325	−21.391	0.003	−21.388	−0.05172	0.00001	−0.05171	−197.06	−0.4162	1.161	−89
−88	2.634	9.351	0.000	9.351	−21.150	0.003	−21.147	−0.05107	0.00001	−0.05106	−196.67	−0.4152	1.259	−88
−87	2.852	9.376	0.000	9.376	−20.909	0.003	−20.906	−0.05042	0.00001	−0.05041	−196.29	−0.4142	1.363	−87
−86	3.089	9.401	0.000	9.401	−20.669	0.003	−20.666	−0.04978	0.00001	−0.04977	−195.90	−0.4131	1.476	−86
−85	3.342	9.426	0.000	9.426	−20.428	0.003	−20.425	−0.04913	0.00001	−0.04912	−195.51	−0.4121	1.597	−85
−84	3.615	9.451	0.000	9.451	−20.188	0.004	−20.184	−0.04849	0.00001	−0.04848	−195.12	−0.4110	1.728	−84
−83	3.909	9.477	0.000	9.477	−19.947	0.004	−19.943	−0.04785	0.00001	−0.04784	−194.73	−0.4100	1.868	−83
−82	4.225	9.502	0.000	9.502	−19.706	0.004	−19.702	−0.04721	0.00001	−0.04720	−194.34	−0.4090	2.019	−82
−81	4.564	9.527	0.000	9.527	−19.466	0.005	−19.461	−0.04658	0.00001	−0.04657	−193.95	−0.4079	2.181	−81
−80	4.930	9.553	0.000	9.553	−19.225	0.005	−19.220	−0.04595	0.00001	−0.04594	−193.55	−0.4069	2.356	−80
−79	5.322	9.578	0.000	9.578	−18.984	0.005	−18.979	−0.04532	0.00002	−0.04530	−193.16	−0.4059	2.543	−79
−78	5.742	9.604	0.000	9.604	−18.744	0.006	−18.738	−0.04469	0.00002	−0.04467	−192.76	−0.4048	2.744	−78
−77	6.193	9.629	0.000	9.629	−18.503	0.006	−18.497	−0.04406	0.00002	−0.04404	−192.37	−0.4038	2.960	−77
−76	6.677	9.654	0.000	9.654	−18.263	0.007	−18.256	−0.04343	0.00002	−0.04341	−191.97	−0.4027	3.192	−76
−75	7.196	9.680	0.000	9.680	−18.022	0.007	−18.015	−0.04280	0.00002	−0.04278	−191.57	−0.4017	3.441	−75
−74	7.753	9.705	0.000	9.705	−17.782	0.008	−17.774	−0.04218	0.00003	−0.04215	−191.17	−0.4007	3.707	−74
−73	8.349	9.730	0.000	9.730	−17.541	0.008	−17.533	−0.04155	0.00003	−0.04152	−190.77	−0.3996	3.992	−73
−72	8.990	9.756	0.000	9.756	−17.301	0.009	−17.292	−0.04093	0.00003	−0.04090	−190.37	−0.3986	4.298	−72
−71	9.675	9.781	0.000	9.781	−17.060	0.010	−17.050	−0.04031	0.00003	−0.04028	−189.97	−0.3975	4.625	−71
−70	10.40	9.806	0.000	9.806	−16.820	0.011	−16.809	−0.03969	0.00003	−0.03966	−189.56	−0.3965	4.976	−70
−69	11.19	9.831	0.000	9.831	−16.579	0.011	−16.568	−0.03908	0.00004	−0.03904	−189.16	−0.3954	5.351	−69
−68	12.03	9.856	0.000	9.856	−16.339	0.012	−16.327	−0.03846	0.00004	−0.03842	−188.75	−0.3944	5.752	−68
−67	12.92	9.882	0.000	9.882	−16.098	0.013	−16.085	−0.03785	0.00004	−0.03781	−188.35	−0.3934	6.181	−67
−66	13.88	9.907	0.000	9.907	−15.858	0.014	−15.844	−0.03724	0.00004	−0.03720	−187.94	−0.3924	6.640	−66
−65	14.91	9.932	0.000	9.932	−15.617	0.015	−15.602	−0.03663	0.00005	−0.03658	−187.53	−0.3913	7.130	−65

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^5$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg $p_s \times 10^3$	
−64	1.601	9.958	0.000	9.958	−15.377	0.016	−15.361	−0.03602	0.00005	−0.03597	−187.12	−0.3903	0.7654	−64
−63	1.718	9.983	0.000	9.983	−15.137	0.018	−15.119	−0.03541	0.00005	−0.03536	−186.71	−0.3893	0.8213	−63
−62	1.843	10.009	0.000	10.009	−14.896	0.019	−14.877	−0.03481	0.00006	−0.03475	−186.30	−0.3882	0.8810	−62
−61	1.976	10.034	0.000	10.034	−14.656	0.020	−14.636	−0.03420	0.00006	−0.03414	−185.89	−0.3872	0.9447	−61
−60	2.118	10.059	0.000	10.059	−14.416	0.022	−14.394	−0.03360	0.00006	−0.03354	−185.47	−0.3861	1.0127	−60
−59	2.269	10.085	0.000	10.085	−14.175	0.023	−14.152	−0.03300	0.00007	−0.03293	−185.06	−0.3851	1.0852	−59
−58	2.431	10.110	0.000	10.110	−13.935	0.025	−13.910	−0.03240	0.00007	−0.03233	−184.64	−0.3841	1.1624	−58
−57	2.603	10.135	0.000	10.135	−13.695	0.027	−13.668	−0.03180	0.00008	−0.03172	−184.23	−0.3830	1.2447	−57
−56	2.786	10.161	0.000	10.161	−13.454	0.029	−13.425	−0.03120	0.00008	−0.03112	−183.81	−0.3820	1.3324	−56
−55	2.982	10.186	0.000	10.186	−13.214	0.031	−13.183	−0.03061	0.00009	−0.03052	−183.39	−0.3810	1.4258	−55
−54	3.190	10.211	0.000	10.211	−12.974	0.033	−12.941	−0.03002	0.00009	−0.02993	−182.97	−0.3799	1.5253	−54
−53	3.411	10.237	0.000	10.237	−12.733	0.035	−12.698	−0.02943	0.0010	−0.02933	−182.55	−0.3789	1.6312	−53
−52	3.646	10.262	0.001	10.263	−12.493	0.038	−12.455	−0.02884	0.00011	−0.02873	−182.13	−0.3779	1.7438	−52
−51	3.897	10.288	0.001	10.289	−12.253	0.041	−12.212	−0.02825	0.00011	−0.02814	−181.71	−0.3769	1.8635	−51
−50	4.163	10.313	0.001	10.314	−12.012	0.043	−11.969	−0.02766	0.00012	−0.02754	−181.29	−0.3758	1.9910	−50
−49	4.446	10.338	0.001	10.339	−11.772	0.046	−11.726	−0.02707	0.00012	−0.02695	−180.87	−0.3748	2.1264	−49
−48	4.747	10.364	0.001	10.365	−11.532	0.049	−11.483	−0.02649	0.00013	−0.02636	−180.44	−0.3738	2.2702	−48
−47	5.066	10.389	0.001	10.390	−11.292	0.053	−11.239	−0.02590	0.00013	−0.02577	−180.02	−0.3728	2.4230	−47
−46	5.406	10.414	0.001	10.415	−11.051	0.056	−10.995	−0.02532	0.00014	−0.02518	−179.59	−0.3717	2.5854	−46
−45	5.766	10.440	0.001	10.441	−10.811	0.060	−10.751	−0.02474	0.00015	−0.02459	−179.16	−0.3707	2.7578	−45
−44	6.149	10.465	0.001	10.466	−10.571	0.064	−10.507	−0.02416	0.00016	−0.02400	−178.73	−0.3696	2.9408	−44
−43	6.555	10.490	0.001	10.491	−10.330	0.068	−10.262	−0.02358	0.00017	−0.02341	−178.30	−0.3686	3.1349	−43
−42	6.985	10.516	0.001	10.517	−10.090	0.073	−10.017	−0.02301	0.00019	−0.02282	−177.87	−0.3676	3.3408	−42
−41	7.441	10.541	0.001	10.542	−9.850	0.078	−9.772	−0.02243	0.00020	−0.02223	−177.44	−0.3665	3.5591	−41
−40	7.925	10.566	0.001	10.567	−9.609	0.083	−9.526	−0.02186	0.00021	−0.02165	−177.01	−0.3655	3.7906	−40
−39	8.437	10.592	0.001	10.593	−9.369	0.089	−9.280	−0.02129	0.00022	−0.02107	−176.58	−0.3645	4.0359	−39
−38	8.980	10.617	0.002	10.619	−9.129	0.094	−9.035	−0.02072	0.00024	−0.02048	−176.14	−0.3634	4.2958	−38
−37	9.556	10.642	0.002	10.644	−8.889	0.100	−8.789	−0.02015	0.00025	−0.01990	−175.71	−0.3624	4.5711	−37
−36	10.16	10.668	0.002	10.670	−8.648	0.106	−8.542	−0.01958	0.00026	−0.01932	−175.27	−0.3614	4.8626	−36
−35	10.81	10.693	0.002	10.695	−8.408	0.113	−8.295	−0.01902	0.00028	−0.01874	−174.84	−0.3604	5.1713	−35
−34	11.49	10.718	0.002	10.720	−8.168	0.121	−8.047	−0.01845	0.00030	−0.01815	−174.40	−0.3593	5.4980	−34
−33	12.21	10.744	0.002	10.746	−7.927	0.128	−7.799	−0.01789	0.00032	−0.01757	−173.96	−0.3583	5.8437	−33

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP. t(F)	HUMIDITY RATIO $W_s \times 10^4$	VOLUME CU FT/LB DRY AIR			ENTHALPY BTU/LB DRY AIR			ENTROPY BTU PER (°F) (LB DRY AIR)			CONDENSED WATER			FAHR. TEMP. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg $p_s \times 10^2$	
−32	1.298	10.769	0.002	10.771	−7.687	0.136	−7.551	−0.01733	0.00034	−0.01699	−173.52	−0.3573	0.62093	−32
−31	1.378	10.794	0.002	10.796	−7.447	0.145	−7.302	−0.01677	0.00036	−0.01641	−173.08	−0.3563	0.65979	−31
−30	1.464	10.820	0.002	10.822	−7.207	0.154	−7.053	−0.01621	0.00038	−0.01583	−172.64	−0.3552	0.70046	−30
−29	1.554	10.845	0.003	10.848	−6.966	0.163	−6.803	−0.01565	0.00040	−0.01525	−172.20	−0.3542	0.74365	−29
−28	1.649	10.870	0.003	10.873	−6.726	0.173	−6.553	−0.01509	0.00043	−0.01466	−171.75	−0.3531	0.78928	−28
−27	1.750	10.896	0.003	10.899	−6.486	0.184	−6.302	−0.01453	0.00045	−0.01408	−171.31	−0.3521	0.83748	−27
−26	1.856	10.921	0.003	10.924	−6.246	0.196	−6.050	−0.01398	0.00048	−0.01350	−170.86	−0.3511	0.88838	−26
−25	1.969	10.946	0.004	10.950	−6.005	0.207	−5.798	−0.01342	0.00051	−0.01291	−170.42	−0.3500	0.94212	−25
−24	2.087	10.972	0.004	10.976	−5.765	0.219	−5.546	−0.01287	0.00054	−0.01233	−169.97	−0.3490	0.99885	−24
−23	2.212	10.997	0.004	11.001	−5.525	0.232	−5.293	−0.01232	0.00057	−0.01175	−169.52	−0.3480	1.0587	−23
−22	2.344	11.022	0.004	11.026	−5.285	0.246	−5.039	−0.01177	0.00061	−0.01116	−169.07	−0.3469	1.1219	−22
−21	2.483	11.048	0.004	11.052	−5.044	0.261	−4.783	−0.01122	0.00064	−0.01058	−168.62	−0.3459	1.1885	−21
−20	2.630	11.073	0.005	11.078	−4.804	0.277	−4.527	−0.01067	0.00068	−0.00999	−168.17	−0.3449	1.2587	−20
−19	2.785	11.098	0.005	11.103	−4.564	0.293	−4.271	−0.01012	0.00072	−0.00940	−167.72	−0.3439	1.3327	−19
−18	2.948	11.124	0.005	11.129	−4.324	0.310	−4.014	−0.00958	0.00076	−0.00882	−167.26	−0.3428	1.4107	−18
−17	3.120	11.149	0.006	11.155	−4.083	0.328	−3.755	−0.00904	0.00080	−0.00824	−166.81	−0.3418	1.4929	−17
−16	3.301	11.174	0.0006	11.180	−3.843	0.348	−3.495	−0.00850	0.00084	−0.00766	−166.35	−0.3408	1.5795	−16
−15	3.491	11.200	0.006	11.206	−3.603	0.368	−3.235	−0.00796	0.00089	−0.00707	−165.90	−0.3398	1.6706	−15
−14	3.692	11.225	0.007	11.232	−3.363	0.389	−2.974	−0.00743	0.00094	−0.00649	−165.44	−0.3387	1.7666	−14
−13	3.903	11.250	0.007	11.257	−3.123	0.412	−2.711	−0.00689	0.00099	−0.00590	−164.98	−0.3377	1.8677	−13
−12	4.125	11.275	0.008	11.283	−2.882	0.436	−2.446	−0.00636	0.00104	−0.00532	−164.52	−0.3367	1.9740	−12
−11	4.359	11.301	0.008	11.309	−2.642	0.461	−2.181	−0.00582	0.00109	−0.00473	−164.06	−0.3357	2.0859	−11
−10	4.606	11.326	0.008	11.334	−2.402	0.487	−1.915	−0.00529	0.00115	−0.00414	−163.60	−0.3346	2.2035	−10
−9	4.865	11.351	0.008	11.359	−2.162	0.514	−1.648	−0.00475	0.00121	−0.00354	−163.14	−0.3336	2.3272	−9
−8	5.137	11.376	0.009	11.385	−1.922	0.543	−1.379	−0.00422	0.00128	−0.00294	−162.67	−0.3326	2.4573	−8
−7	5.423	11.401	0.010	11.411	−1.681	0.574	−1.107	−0.00369	0.00135	−0.00234	−162.21	−0.3316	2.5940	−7
−6	5.724	11.427	0.010	11.437	−1.441	0.606	−0.835	−0.00316	0.00142	−0.00174	−161.74	−0.3305	2.7377	−6
−5	6.040	11.452	0.011	11.463	−1.201	0.639	−0.562	−0.00263	0.00149	−0.00114	−161.28	−0.3295	2.8886	−5
−4	6.371	11.477	0.012	11.489	−0.961	0.675	−0.286	−0.00210	0.00157	−0.00053	−160.81	−0.3285	3.0472	−4
−3	6.720	11.502	0.013	11.515	−0.721	0.712	−0.009	−0.00157	0.00165	0.00008	−160.34	−0.3275	3.2137	−3
−2	7.085	11.528	0.013	11.541	−0.480	0.751	0.271	−0.00105	0.00174	0.00069	−159.87	−0.3264	3.3885	−2
−1	7.469	11.553	0.014	11.567	−0.240	0.792	0.552	−0.00052	0.00183	0.00131	−159.40	−0.3254	3.5720	−1

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^3$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg $p_s \times 10^2$	
0	0.7872	11.578	0.015	11.593	0.000	0.835	0.835	0.00000	0.00192	0.00192	−158.93	−0.3244	3.7645	0
1	0.8295	11.604	0.015	11.619	0.240	0.880	1.120	0.00052	0.00202	0.00254	−158.46	−0.3234	3.9666	1
2	0.8739	11.629	0.016	11.645	0.480	0.928	1.408	0.00104	0.00212	0.00316	−157.99	−0.3223	4.1785	2
3	0.9204	11.654	0.017	11.671	0.721	0.977	1.698	0.00156	0.00223	0.00379	−157.52	−0.3213	4.4007	3
4	0.9692	11.679	0.018	11.697	0.961	1.030	1.991	0.00208	0.00234	0.00442	−157.04	−0.3203	4.6337	4
5	1.020	11.705	0.019	11.724	1.201	1.085	2.286	0.00260	0.00246	0.00506	−156.57	−0.3193	4.8779	5
6	1.074	11.730	0.020	11.750	1.441	1.142	2.583	0.00312	0.00258	0.00570	−156.09	−0.3182	5.1339	6
7	1.130	11.756	0.021	11.777	1.681	1.202	2.883	0.00364	0.00271	0.00635	−155.61	−0.3172	5.4022	7
8	1.189	11.781	0.022	11.803	1.922	1.266	3.188	0.00415	0.00285	0.00700	−155.13	−0.3162	5.6832	8
9	1.251	11.806	0.024	11.830	2.162	1.332	3.494	0.00467	0.00299	0.00766	−154.65	−0.3152	5.9776	9
10	1.315	11.831	0.025	11.856	2.402	1.401	3.803	0.00518	0.00314	0.00832	−154.17	−0.3141	6.2858	10
11	1.383	11.857	0.026	11.883	2.642	1.474	4.116	0.00569	0.00330	0.00899	−153.69	−0.3131	6.6085	11
12	1.454	11.882	0.028	11.910	2.882	1.550	4.432	0.00620	0.00346	0.00966	−153.21	−0.3121	6.9462	12
13	1.528	11.907	0.029	11.936	3.123	1.630	4.753	0.00671	0.00363	0.01034	−152.73	−0.3111	7.2997	13
14	1.606	11.933	0.030	11.963	3.363	1.713	5.076	0.00721	0.00380	0.01101	−152.24	−0.3100	7.6696	14
15	1.687	11.958	0.032	11.990	3.603	1.800	5.403	0.00772	0.00399	0.01171	−151.76	−0.3090	8.0565	15
16	1.772	11.983	0.034	12.017	3.843	1.892	5.735	0.00822	0.00418	0.01240	−151.27	−0.3080	8.4612	16
17	1.861	12.009	0.035	12.044	4.083	1.988	6.071	0.00873	0.00438	0.01311	−150.78	−0.3070	8.8843	17
18	1.953	12.034	0.038	12.072	4.324	2.088	6.412	0.00923	0.00459	0.01382	−150.29	−0.3059	9.3267	18
19	2.051	12.059	0.040	12.099	4.564	2.192	6.756	0.00973	0.00481	0.01454	−149.80	−0.3049	9.7889	19
20	2.152	12.084	0.042	12.126	4.804	2.302	7.106	0.01023	0.00504	0.01527	−149.31	−0.3039	10.272	20
21	2.258	12.110	0.044	12.154	5.044	2.416	7.460	0.01073	0.00528	0.01601	−148.82	−0.3029	10.777	21
22	2.369	12.135	0.046	12.181	5.284	2.536	7.820	0.01123	0.00553	0.01676	−148.33	−0.3018	11.305	22
23	2.485	12.160	0.049	12.209	5.525	2.661	8.186	0.01173	0.00579	0.01752	−147.84	−0.3008	11.856	23
24	2.606	12.186	0.051	12.237	5.765	2.792	8.557	0.01223	0.00607	0.01830	−147.34	−0.2998	12.431	24
25	2.733	12.211	0.054	12.265	6.005	2.929	8.934	0.01273	0.00635	0.01908	−146.85	−0.2988	13.032	25
26	2.865	12.236	0.057	12.293	6.245	3.072	9.317	0.01322	0.00665	0.01987	−146.35	−0.2977	13.659	26
27	3.003	12.262	0.059	12.321	6.485	3.221	9.706	0.01372	0.00696	0.02068	−145.85	−0.2967	14.313	27
28	3.147	12.287	0.062	12.349	6.726	3.377	10.103	0.01421	0.00728	0.02149	−145.36	−0.2957	14.996	28
29	3.297	12.312	0.065	12.377	6.966	3.540	10.506	0.01470	0.00761	0.02231	−144.86	−0.2947	15.709	29
30	3.454	12.338	0.068	12.406	7.206	3.709	10.915	0.01519	0.00796	0.02315	−144.36	−0.2936	16.452	30
31	3.617	12.363	0.071	12.434	7.446	3.887	11.333	0.01568	0.00832	0.02400	−143.86	−0.2926	17.227	31
32	3.788	12.388	0.075	12.463	7.686	4.072	11.758	0.01617	0.00870	0.02487	−143.36	−0.2916	18.035	32
*32	3.788	12.388	0.075	12.463	7.686	4.072	11.758	0.01617	0.00870	0.02487	0.04	0.0000	18.037	32*
33	3.944	12.413	0.079	12.492	7.927	4.242	12.169	0.01666	0.00904	0.02570	1.05	0.0020	18.778	33
34	4.107	12.438	0.082	12.520	8.167	4.418	12.585	0.01715	0.00940	0.02655	2.06	0.0041	19.546	34

[a]Compiled by John A. Goff and S. Gratch.
*Extrapolated to represent metastable equilibrium with undercooled liquid

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP. t(F)	HUMIDITY RATIO $W_s \times 10^3$	VOLUME CU FT/LB DRY AIR			ENTHALPY BTU/LB DRY AIR			ENTROPY BTU PER (°F) (LB DRY AIR)			CONDENSED WATER			FAHR. TEMP. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb) (°F) s_w	Vap. Press In. Hg p_s	
35	4.275	12.464	0.085	12.549	8.407	4.601	13.008	0 01764	0 00977	0 02741	3.06	0 00061	0.20342	35
36	4 450	12 489	0 089	12 578	8 647	4.791	13.438	0.01812	0 01016	0 02828	4 07	0 0081	0 21166	36
37	4.631	12.514	0.093	12.607	8.887	4.987	13.874	0.01861	0.01056	0.02917	5.07	0.0102	0.22020	37
38	4 818	12 540	0.097	12.637	9.128	5.191	14 319	0 01909	0 01097	0.03006	6.08	0 0122	0 22904	38
39	5.012	12.565	0.101	12.666	9.368	5.403	14.771	0.01957	0.01139	0.03096	7.08	0.0142	0.23819	39
40	5 213	12.590	0.105	12 695	9.608	5.622	15.230	0.02005	0.01183	0.03188	8.09	0.0162	0 24767	40
41	5.421	12.616	0.109	12.725	9.848	5.849	15 697	0 02053	0 01228	0 03281	9 09	0 0182	0 25748	41
42	5 638	12.641	0.114	12 755	10 088	6.084	16.172	0.02101	0 01275	0.03376	10.09	0.0202	0.26763	42
43	5.860	12.666	0.119	12.785	10 329	6.328	16.657	0 02149	0 01323	0 03472	11.10	0.0222	0 27813	43
44	6.091	12.691	0.124	12.815	10 569	6.580	17.149	0 02197	0 01373	0.03570	12.10	0.0242	0.28899	44
45	6 331	12.717	0.129	12 846	10.809	6.841	17.650	0.02245	0.01425	0.03670	13.10	0.0262	0.30023	45
46	6 578	12.742	0.134	12.876	11.049	7.112	18.161	0.02293	0 01478	0.03771	14.10	0 0282	0.31185	46
47	6.835	12.767	0.140	12.907	11.289	7.391	18.680	0.02340	0.01534	0.03874	15 11	0.0302	0 32386	47
48	7.100	12.792	0.146	12.938	11.530	7.681	19.211	0 02387	0.01591	0 03978	16.11	0.0321	0.33629	48
49	7 374	12.818	0.151	12.969	11.770	7.981	19.751	0.02434	0.01650	0 04084	17 11	0.0341	0 34913	49
50	7.658	12.843	0.158	13.001	12.010	8.291	20.301	0.02481	0 01711	0 04192	18.11	0.0361	0 36240	50
51	7 952	12 868	0 164	13 032	12.250	8.612	20 862	0 02528	0 01774	0 04302	19.11	0.0381	0.37611	51
52	8.256	12 894	0.170	13.064	12 491	8 945	21.436	0 02575	0.01839	0.04414	20.11	0.0400	0.39028	52
53	8 569	12.919	0.178	13.097	12.731	9.289	22.020	0.02622	0.01906	0.04528	21.12	0.0420	0.40492	53
54	8.894	12.944	0.185	13.129	12.971	9.644	22.615	0 02669	0 01976	0.04645	22.12	0.0439	0.42004	54
55	9.229	12.970	0 192	13.162	13 211	10.01	23.22	0 02716	0 02047	0.04763	23.12	0.0459	0.43565	55
56	9.575	12.995	0.200	13.195	13 452	10.39	23.84	0 02762	0 02121	0 04883	24.12	0.0478	0.45176	56
57	9.934	13.020	0.208	13 228	13 692	10.79	24.48	0.02809	0.02197	0 05006	25 12	0 0497	0.46840	57
58	10.30	13.045	0.216	13.261	13.932	11.19	25.12	0.02855	0.02276	0 05131	26.12	0.0517	0 48558	58
59	10.69	13 071	0.224	13.295	14 172	11.61	25 78	0.02902	0 02357	0.05259	27.12	0.0536	0.50330	59
60	11.08	13 096	0 233	13 329	14.413	12.05	26.46	0.02948	0 02441	0.05389	28 12	0.0555	0.52159	60
61	11.49	13.121	0.242	13.363	14.653	12.50	27.15	0.02994	0.02527	0.05521	29.12	0 0574	0 54047	61
62	11.91	13.147	0.251	13 398	14.893	12.96	27.85	0 03040	0.02616	0.05656	30.12	0.0594	0.55994	62
63	12.35	13.172	0.261	13.433	15 134	13.44	28.57	0.03086	0.02708	0.05794	31.12	0.0613	0.58002	63
64	12.80	13 197	0 271	13.468	15.374	13.94	29.31	0 03132	0.02803	0.05935	32.12	0.0632	0.60073	64
65	13.26	13.222	0.282	13.504	15.614	14.45	30.06	0 03177	0 02901	0 06078	33.11	0.0651	0.62209	65
66	13.74	13.247	0 292	13.539	15.855	14.98	30.83	0.03223	0.03002	0.06225	34.11	0.0670	0.64411	66
67	14.24	13.273	0.303	13.576	16.095	15.53	31.62	0 03269	0 03106	0.06375	35 11	0 0689	0.66681	67
68	14 75	13.298	0.315	13.613	16.335	16.09	32.42	0.03314	0.03213	0.06527	36.11	0.0708	0.69019	68
69	15.28	13.323	0.327	13.650	16 576	16 67	33 25	0 03360	0.03323	0.06683	37.11	0.0727	0.71430	69

[a]Compiled by John A Goff and S Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^2$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In Hg p_s	
70	1.582	13.348	0.339	13.687	16.816	17.27	34.09	0.03405	0.03437	0.06842	38.11	0.0746	0.73915	70
71	1.639	13.373	0.351	13.724	17.056	17.89	34.95	0.03450	0.03554	0.07004	39.11	0.0765	0.76475	71
72	1.697	13.398	0.364	13.762	17.297	18.53	35.83	0.03495	0.03675	0.07170	40.11	0.0784	0.79112	72
73	1.757	13.424	0.377	13.801	17.537	19.20	36.74	0.03540	0.03800	0.07340	41.11	0.0803	0.81828	73
74	1.819	13.449	0.392	13.841	17.778	19.88	37.66	0.03585	0.03928	0.07513	42.10	0.0821	0.84624	74
75	1.882	13.474	0.407	13.881	18.018	20.59	38.61	0.03630	0.04060	0.07690	43.10	0.0840	0.87504	75
76	1.948	13.499	0.422	13.921	18.259	21.31	39.57	0.03675	0.04197	0.07872	44.10	0.0859	0.90470	76
77	2.016	13.525	0.437	13.962	18.499	22.07	40.57	0.03720	0.04337	0.08057	45.10	0.0877	0.93523	77
78	2.086	13.550	0.453	14.003	18.740	22.84	41.58	0.03765	0.04482	0.08247	46.10	0.0896	0.96665	78
79	2.158	13.575	0.470	14.045	18.980	23.64	42.62	0.03810	0.04631	0.08441	47.10	0.0914	0.99899	79
80	2.233	13.601	0.486	14.087	19.221	24.47	43.69	0.03854	0.04784	0.08638	48.10	0.0933	1.0323	80
81	2.310	13.626	0.504	14.130	19.461	25.32	44.78	0.03899	0.04942	0.08841	49.09	0.0952	1.0665	81
82	2.389	13.651	0.523	14.174	19.702	26.20	45.90	0.03943	0.05105	0.09048	50.09	0.0970	1.1017	82
83	2.471	13.676	0.542	14.218	19.942	27.10	47.04	0.03987	0.05273	0.09260	51.09	0.0989	1.1379	83
84	2.555	13.702	0.560	14.262	20.183	28.04	48.22	0.04031	0.05446	0.09477	52.09	0.1007	1.1752	84
85	2.642	13.727	0.581	14.308	20.423	29.01	49.43	0.04075	0.05624	0.09699	53.09	0.1025	1.2135	85
86	2.731	13.752	0.602	14.354	20.663	30.00	50.66	0.04119	0.05807	0.09926	54.08	0.1043	1.2529	86
87	2.824	13.777	0.624	14.401	20.904	31.03	51.93	0.04163	0.05995	0.10158	55.08	0.1062	1.2934	87
88	2.919	13.803	0.645	14.448	21.144	32.09	53.23	0.04207	0.06189	0.10396	56.08	0.1080	1.3351	88
89	3.017	13.828	0.668	14.496	21.385	33.18	54.56	0.04251	0.06389	0.10640	57.08	0.1098	1.3779	89
90	3.118	13.853	0.692	14.545	21.625	34.31	55.93	0.04295	0.06596	0.10890	58.08	0.1116	1.4219	90
91	3.223	13.879	0.716	14.595	21.865	35.47	57.33	0.04339	0.06807	0.11146	59.07	0.1135	1.4671	91
92	3.330	13.904	0.741	14.645	22.106	36.67	58.78	0.04382	0.07025	0.11407	60.07	0.1153	1.5135	92
93	3.441	13.929	0.768	14.697	22.346	37.90	60.25	0.04426	0.07249	0.11675	61.07	0.1171	1.5612	93
94	3.556	13.954	0.795	14.749	22.587	39.18	61.77	0.04469	0.07480	0.11949	62.07	0.1188	1.6102	94
95	3.673	13.980	0.822	14.802	22.827	40.49	63.32	0.04513	0.07718	0.12231	63.07	0.1206	1.6606	95
96	3.795	14.005	0.851	14.856	23.068	41.85	64.92	0.04556	0.07963	0.12519	64.06	0.1224	1.7123	96
97	3.920	14.030	0.881	14.911	23.308	43.24	66.55	0.04600	0.08215	0.12815	65.06	0.1242	1.7654	97
98	4.049	14.056	0.911	14.967	23.548	44.68	68.23	0.04643	0.08474	0.13117	66.06	0.1260	1.8199	98
99	4.182	14.081	0.942	15.023	23.789	46.17	69.96	0.04686	0.08741	0.13427	67.06	0.1278	1.8759	99
100	4.319	14.106	0.975	15.081	24.029	47.70	71.73	0.04729	0.09016	0.13745	68.06	0.1296	1.9333	100
101	4.460	14.131	1.009	15.140	24.270	49.28	73.55	0.04772	0.09299	0.14071	69.05	0.1314	1.9923	101
102	4.608	14.157	1.043	15.200	24.510	50.91	75.42	0.04815	0.09591	0.14406	70.05	0.1332	2.0528	102
103	4.756	14.182	1.079	15.261	24.751	52.59	77.34	0.04858	0.09891	0.14749	71.05	0.1350	2.1149	103
104	4.911	14.207	1.117	15.324	24.991	54.32	79.31	0.04900	0.1020	0.1510	72.05	0.1367	2.1786	104

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TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP. t(F)	HUMIDITY RATIO W_s x 10	VOLUME CU FT/LB DRY AIR			ENTHALPY BTU/LB DRY AIR			ENTROPY BTU PER (°F) (LB DRY AIR)			CONDENSED WATER			FAHR TEMP. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In Hg p_s	
105	0.5070	14.232	1.155	15.387	25.232	56.11	81.34	0.04943	0 1052	0.1546	73 04	0.1385	2.2439	105
106	0.5234	14.258	1.194	15 452	25.472	57 95	83.42	0.04985	0 1085	0 1584	74.04	0 1403	2.3109	106
107	0.5404	14 283	1.235	15 518	25.713	59 85	85 56	0.05028	0.1118	0.1621	75.04	0.1421	2.3797	107
108	0.5578	14 308	1 278	15.586	25 953	61 80	87 76	0 05070	0.1153	0.1660	76.04	0.1438	2.4502	108
109	0.5758	14.333	1.321	15.654	26.194	63.82	90.03	0.05113	0.1189	0.1700	77.04	0.1456	2.5225	109
110	0.5944	14.359	1.365	15.724	26.434	65.91	92.34	0.05155	0 1226	0.1742	78.03	0.1472	2.5966	110
111	0.6135	14.384	1 412	15 796	26.675	68 05	94.72	0 05197	0.1264	0.1784	79.03	0.1491	2.6726	111
112	0.6333	14.409	1.460	15.869	26.915	70.27	97.18	0.05239	0 1302	0 1826	80.03	0.1508	2 7505	112
113	0.6536	14.435	1.509	15 944	27.156	72.55	99.71	0 05281	0 1342	0 1870	81 03	0.1525	2 8304	113
114	0.6746	14 460	1.560	16.020	27.397	74.91	102.31	0.05323	0.1384	0 1916	82.03	0.1543	2 9123	114
115	0.6962	14.485	1.613	16 098	27 637	77.34	104.98	0.05365	0.1426	0.1963	83 02	0.1560	2 9962	115
116	0.7185	14.510	1.668	16 178	27.878	79.85	107.73	0 05407	0 1470	0 2011	84 02	0 1577	3 0821	116
117	0.7415	14 536	1.723	16 259	28.119	82.43	110 55	0 05449	0 1515	0.2060	85 02	0.1595	3.1701	117
118	0.7652	14.561	1.782	16.343	28 359	85.10	113.46	0.05490	0 1562	0.2111	86.02	0.1612	3.2603	118
119	0 7897	14 586	1.842	16.428	28 600	87.86	116.46	0.05532	0.1610	0.2163	87 02	0.1629	3.3527	119
120	0 8149	14.611	1.905	16.516	28 841	90.70	119.54	0.05573	0 1659	0 2216	88.01	0 1646	3 4474	120
121	0 8410	14.637	1.968	16 605	29 082	93.64	122.72	0.05615	0 1710	0 2272	89.01	0 1664	3.5443	121
122	0 8678	14.662	2.034	16 696	29 322	96 66	125 98	0 05656	0 1763	0 2329	90.01	0.1681	3.6436	122
123	0 8955	14 687	2 103	16 790	29 563	99.79	129.35	0.05698	0.1817	0.2387	91.01	0.1698	3 7452	123
124	0.9242	14.712	2.174	16.886	29.804	103.0	132.8	0.05739	0.1872	0 2446	92.01	0.1715	3.8493	124
125	0 9537	14.738	2.247	16.985	30.044	106.4	136.4	0.05780	0.1930	0.2508	93 01	0 1732	3.9558	125
126	0 9841	14 763	2.323	17 086	30 285	109 8	140 1	0 05821	0.1989	0 2571	94.01	0.1749	4 0649	126
127	1 016	14 788	2 401	17 189	30.526	113.4	143.9	0 05862	0.2050	0 2636	95 00	0.1766	4.1765	127
128	1 048	14 813	2.482	17.295	30.766	117.0	147 8	0 05903	0 2113	0 2703	96.00	0.1783	4.2907	128
129	1.082	14.839	2 565	17.404	31 007	120 8	151 8	0 05944	0.2178	0 2772	97.00	0.1800	4.4076	129
130	1.116	14.864	2.652	17.516	31 248	124.7	155.9	0 05985	0 2245	0.2844	98 00	0 1817	4 5272	130
131	1.152	14.889	2.742	17 631	31.489	128 8	160 3	0 06026	0 2314	0.2917	99.00	0.1834	4.6495	131
132	1.189	14.915	2 834	17 749	31 729	133 0	164.7	0.06067	0 2386	0 2993	100 00	0 1851	4 7747	132
133	1.227	14 940	2.930	17.870	31.970	137.3	169 3	0 06108	0.2459	0 3070	101 00	0 1868	4 9028	133
134	1.267	14.965	3.029	17.994	32.211	141.8	174.0	0 06148	0 2536	0 3151	102.00	0.1885	5.0337	134
135	1.308	14.990	3.132	18 122	32.452	146.4	178 9	0 06189	0 2614	0.3233	103.00	0.1902	5.1676	135
136	1.350	15 016	3.237	18 253	32.692	151.2	183.9	0 06229	0.2695	0.3318	104.00	0.1918	5 3046	136
137	1.393	15 041	3.348	18 389	32.933	156.1	189.0	0 06270	0.2778	0 3405	105 00	0 1935	5.4446	137
138	1.439	15 066	3.462	18 528	33.174	161 2	194.4	0.06310	0.2865	0.3496	106.00	0.1952	5.5878	138
139	1.485	15 091	3.580	18.671	33.414	166.5	199.9	0.06350	0.2954	0.3589	107.00	0 1969	5.7342	139

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP. t(F)	HUMIDITY RATIO W_s	VOLUME CU FT/LB DRY AIR			ENTHALPY BTU/LB DRY AIR			ENTROPY BTU PER (°F) (LB DRY AIR)			CONDENSED WATER			FAHR. TEMP. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg p_s	
140	0.1534	15.117	3.702	18.819	33.655	172.0	205.7	0 06390	0 3047	0 3686	107 99	0 1985	5 8838	140
141	0.1584	15.142	3 829	18.971	33 896	177.7	211 6	0 06430	0 3142	0.3785	108 99	0 2002	6.0367	141
142	0.1636	15.167	3.961	19 128	34.136	183 6	217.7	0 06470	0 3241	0 3888	109.99	0.2018	6 1930	142
143	0.1689	15.192	4 098	19.290	34.377	189.7	224.1	0 06510	0 3343	0.3994	110.99	0 2035	6.3527	143
144	0.1745	15.218	4.239	19.457	34.618	196 0	230 6	0.06549	0.3449	0.4104	111.99	0.2051	6 5160	144
145	0.1803	15 243	4 386	19.629	34.859	202.5	237 4	0.06589	0 3559	0.4218	112.99	0.2068	6 6828	145
146	0.1862	15.268	4.539	19.807	35 099	209.3	244.4	0.06629	0.3672	0.4335	113 99	0.2084	6 8532	146
147	0.1924	15.293	4 698	19.991	35 340	216 4	251.7	0.06669	0 3790	0.4457	114 99	0 2101	7 0273	147
148	0.1989	15.319	4 862	20.181	35 581	223.7	259 3	0 06708	0.3912	0 4583	115 99	0 2117	7.2051	148
149	0 2055	15 344	5.033	20 377	35 822	231.3	267 1	0 06748	0.4038	0.4713	116 99	0.2134	7.3867	149
150	0.2125	15 369	5.211	20 580	36.063	239 2	275.3	0 06787	0.4169	0 4848	117 99	0 2150	7.5722	150
151	0.2197	15 394	5 396	20.790	36 304	247.3	283 6	0 06827	0.4304	0 4987	118 99	0.2167	7 7616	151
152	0 2271	15.420	5 587	21.007	36 545	255.9	292 4	0 06866	0 4445	0 5132	119 99	0.2183	7.9550	152
153	0.2349	15 445	5.788	21.233	36 785	264 7	301 5	0.06906	0.4591	0.5282	120 99	0 2200	8 1525	153
154	0 2430	15.470	5.996	21.466	37 026	273.9	310 9	0 06945	0.4743	0.5438	121.99	0.2216	8 3541	154
155	0.2514	15.496	6 213	21.709	37.267	283.5	320.8	0.06984	0 4901	0.5599	122.99	0 2232	8 5599	155
156	0.2602	15 521	6.439	21 960	37 508	293 5	331.0	0.07023	0.5066	0.5768	123.99	0 2248	8 7701	156
157	0.2693	15.546	6.675	22.221	37.749	303 9	341.7	0 07062	0 5237	0 5943	124.99	0.2265	8.9846	157
158	0.2788	15.571	6.922	22.493	37.990	314.7	352.7	0 07101	0 5415	0.6125	125.99	0 2281	9 2036	158
159	0.2887	15.597	7.178	22.775	38.231	326 0	364 2	0 07140	0.5600	0.6314	127.00	0 2297	9.4271	159
160	0.2990	15.622	7.446	23.068	38.472	337 8	376 3	0 07179	0 5793	0 6511	128 00	0.2313	9.6556	160
161	0.3098	15 647	7 727	23.374	38.713	350 1	388 8	0 07218	0 5994	0 6716	129.00	0.2329	9.8876	161
162	0.3211	15 672	8 020	23 692	38.954	363 0	402.0	0.07257	0 6204	0.6930	130 00	0 2345	10 125	162
163	0.3329	15.698	8.326	24.024	39 195	376 5	415.7	0.07296	0 6423	0 7153	131 00	0.2361	10 367	163
164	0.3452	15 723	8 648	24.371	39.436	390 5	429.9	0.07334	0 6652	0.7385	132.00	0 2377	10.614	164
165	0.3581	15.748	8 985	24.733	39.677	405.3	445.0	0.07373	0.6892	0 7629	133 00	0.2393	10 866	165
166	0 3716	15.773	9.339	25 112	39 918	420.8	460 7	0 07411	0 7142	0 7883	134.00	0 2409	11 123	166
167	0 3858	15.799	9.708	25.507	40.159	437.0	447.2	0 07450	0 7405	0.8150	135.01	0 2426	11 385	167
168	0.4007	15.824	10.098	25 922	40 400	454 0	494.4	0 07488	0.7680	0.8429	136 01	0.2441	11 652	168
169	0.4163	15.849	10.508	26.357	40 641	471.8	512 4	0 07527	0.7969	0 8722	137.01	0.2457	11.925	169
170	0.4327	15.874	10.938	26 812	40.882	490 6	531 5	0.07565	0.8273	0 9030	138.01	0 2473	12.203	170
171	0.4500	15.900	11.391	27 291	41.123	510 4	551.5	0 07603	0 8592	0.9352	139.01	0 2489	12.480	171
172	0.4682	15.925	11.870	27.795	41.364	531 3	572.7	0.07641	0.8927	0 9691	140 01	0 2505	12 775	172
173	0.4875	15.950	12 376	28 326	41.605	553.3	594 9	0 07680	0.9281	1.0049	141.01	0.2521	13.069	173
174	0 5078	15 975	12.911	28.886	41.846	576.5	618 3	0 07718	0 9654	1 0428	142.02	0.2537	13.369	174

[a]Compiled by John A. Goff and S. Gratch.

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Concluded)

Fahr. Temp. t(F)	Humidity Ratio W_s	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg p_s	Fahr. Temp. t(F)
175	0.5292	16 001	13 475	29.476	42.087	601.1	643.2	0.07756	1.005	1.083	143.02	0.2553	13.675	175
176	0 5519	16 026	14.074	30 100	42 328	627 1	66.94	0.07794	1.047	1 125	144.02	0 2568	13.987	176
177	0 5760	16 051	14.710	30 761	42 569	654 7	697 3	0 07832	1.091	1 169	145 02	0.2584	14.304	177
178	0 6016	16 076	15.386	31.462	42 810	684.1	726.9	0.07870	1.137	1 216	146.03	0.2600	14.628	178
179	0.6288	16.102	16.104	32.206	43.051	715.2	758.3	0.07908	1.187	1.266	147.03	0.2616	14.958	179
180	0 6578	16.127	16 870	32.997	43 292	748 5	791 8	0.07946	1.240	1 319	148.03	0.2631	15.294	180
181	0.6887	16.152	17 689	33 841	43.534	783.9	827.4	0.07984	1.296	1.376	149.03	0.2647	15.636	181
182	0.7218	16 177	18 565	34 742	43.775	821 9	865.7	0.08021	1.357	1.437	150 04	0.2662	15.985	182
183	0.7572	16.203	19.504	35 707	44.016	862 5	906 5	0 08059	1.421	1.502	151.04	0.2678	16 340	183
184	0.7953	16.228	20 513	36.741	44.257	906.2	950.5	0 08096	1.490	1.571	152.04	0.2693	16.702	184
185	0.8363	16.253	21 601	37.854	44.498	953.2	997.7	0 08134	1.565	1.646	153.05	0.2709	17.071	185
186	0 8805	16 278	22.775	39.053	44 740	1004	1049	0 08171	1.645	1.727	154.05	0.2724	17.446	186
187	0.9283	16 304	24.047	40.351	44.981	1059	1104	0.08208	1.731	1.813	155.05	0.2740	17.828	187
188	0 9802	16 329	25.427	41.756	45 222	1119	1164	0 08245	1.825	1.907	156.06	0.2755	18.217	188
189	1.037	16.354	26.934	43.288	45.463	1184	1229	0.08283	1 928	2.011	157 06	0 2771	18 614	189
190	1 099	16 379	28.580	44.959	45.704	1255	1301	0 08320	2.039	2 122	158.07	0.2786	19.017	190
191	1.166	16.405	30.385	46.790	45 946	1332	1378	0.08357	2.161	2.245	159.07	0.2802	19.427	191
192	1 241	16 430	32.375	48 805	46 187	1418	1464	0.08394	2.296	2.380	160.07	0.2817	19.845	192
193	1 324	16 455	34.581	51.036	46 428	1513	1559	0.08431	2.444	2.528	161.08	0.2833	20.271	193
194	1 416	16 480	37.036	53.516	46 670	1619	1666	0.08468	2.609	2.694	162.08	0 2848	20.704	194
195	1 519	16 506	39.785	56.291	46 911	1737	1784	0 08505	2.794	2.879	163.09	0 2864	21 145	195
196	1.635	16 531	42.885	59.416	47.153	1871	1918	0 08542	3.002	3.087	164.09	0.2879	21.594	196
197	1.767	16 556	46 402	62.958	47 394	2022	2069	0.08579	3.238	3.324	165.10	0.2895	22 050	197
198	1 917	16 581	50 426	67.007	47.636	2195	2243	0.08616	3.507	3.593	166.10	0.2910	22.514	198
199	2 091	16 607	55 074	71.681	47.877	2395	2443	0.08653	3.817	3.904	167.11	0.2925	22.987	199
200	2.295	16 632	60 510	77 142	48.119	2629	2677	0.08689	4.179	4.266	168.11	0.2940	23.468	200

[a]Compiled by John A. Goff and S. Gratch.

given temperature (below the critical temperature), though this requires that it have a definite pressure p_s and that the coexisting condensed phase have the same temperature and pressure. The values of saturation pressure listed in Table 1 have been computed from the formulae of Goff and Gratch [2].

Thermodynamic Properties of Water at Saturation

Table 2 offers revisions to existing steam table data [3] with extension downward to −160 F. These revisions and extension were a necessary preliminary to the construction of Table 1. A detailed explanation of the methods employed in constructing Table 2 is given in a paper [2] by John A. Goff and S. Gratch. As in Table 1 the temperature scale used as argument in Table 2 is the Fahrenheit scale defined in terms of absolute temperature T by Equation 1, whereas the Fahrenheit scale used as argument in existing steam tables is that derived from the International Centigrade scale by means of Equation 2. The symbols used as column headings in Table 2 are the same as those used in the steam tables and have the same meanings; therefore, a detailed explanation seems unnecessary.

Properties of water above 212 F from Keenan and Keyes [3] are given in Table 3.

DEGREE OF SATURATION

At given values of temperature and pressure the humidity ratio W of moist air can have any value between zero (dry air) and W_s (moist air at saturation). For convenience a parameter μ called alternatively ***degree of saturation*** or ***per cent saturation*** is introduced through the definition,

$$W = \mu W_s \tag{3}$$

Obviously the degree of saturation μ can have any value from zero (dry air) to unity (moist air at saturation).

To a degree of approximation within the estimated uncertainty of the data in Table 1 at temperatures below about 150 F, the volume v of moist air per pound of dry air at any degree of saturation μ may be computed from the simple relation,

$$v = v_a + \mu v_{as} \tag{4}$$

To obtain comparable accuracy at temperatures above about 150 F it is necessary to add a correction term $\overline{v}$ as follows,

$$\overline{v} = \frac{\mu(1 - \mu)A}{1 + aW_s\,\mu} \tag{4a}$$

where a denotes the ratio of the apparent molecular weight of dry air (28.966) to the molecular weight of water (18.016), namely, 1.6078. In Table 4 are given, for each of several higher temperatures, the corresponding value of the coefficient A, the value of μ at which the correction term $\overline{v}$ attains its maximum value, and the maximum value of the correction term there attained.

At temperatures below about 150 F the enthalpy h of moist air per pound of dry air at any degree of saturation μ may be computed from the simple relation,

$$h = h_a + \mu h_{as} \tag{5}$$

To obtain comparable accuracy at temperatures above about 150 F it is

Table 2. Thermodynamic Properties of Water at Saturation[a]

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^7$		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap $v_{ig} \times 10^{-8}$	Sat. Vapor $v_g \times 10^{-8}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−160	0.4949	1.008	0.01722	36.07	36.07	−222.05	1212.43	990.38	−0.4907	4.0456	3.5549	−160
−159	0.5592	1.138	0.01722	32.03	32.03	−221.73	1212.55	990.82	−0.4896	4.0325	3.5429	−159
−158	0.6312	1.285	0.01723	28.47	28.47	−221.41	1212.67	991.26	−0.4886	4.0196	3.5310	−158
−157	0.7121	1.450	0.01723	25.32	25.32	−221.09	1212.79	991.70	−0.4875	4.0067	3.5192	−157
−156	0.8026	1.634	0.01723	22.54	22.54	−220.76	1212.90	992.14	−0.4864	3.9939	3.5075	−156
−155	0.9040	1.840	0.01723	20.08	20.08	−220.44	1213.02	992.58	−0.4854	3.9812	3.4958	−155
−154	1.017	2.072	0.01723	17.90	17.90	−220.12	1213.15	993.03	−0.4843	3.9685	3.4842	−154
−153	1.144	2.329	0.01723	15.97	15.97	−219.79	1213.26	993.47	−0.4832	3.9559	3.4727	−153
−152	1.286	2.618	0.01723	14.26	14.26	−219.47	1213.38	993.91	−0.4822	3.9435	3.4613	−152
−151	1.444	2.939	0.01723	12.74	12.74	−219.14	1213.49	994.35	−0.4811	3.9311	3.4500	−151
−150	1.620	3.298	0.01723	11.39	11.39	−218.82	1213.62	994.80	−0.4801	3.9188	3.4387	−150
−149	1.816	3.698	0.01723	10.19	10.19	−218.49	1213.73	995.24	−0.4790	3.9065	3.4275	−149
−148	2.035	4.143	0.01724	9.123	9.123	−218.16	1213.84	995.68	−0.4780	3.8944	3.4164	−148
−147	2.278	4.639	0.01724	8.174	8.174	−217.83	1213.95	996.12	−0.4769	3.8823	3.4054	−147
−146	2.549	5.190	0.01724	7.330	7.330	−217.50	1214.06	996.56	−0.4758	3.8702	3.3944	−146
−145	2.850	5.803	0.01724	6.577	6.577	−217.17	1214.17	997.00	−0.4748	3.8583	3.3835	−145
−144	3.184	6.483	0.01724	5.905	5.905	−216.83	1214.28	997.45	−0.4737	3.8464	3.3727	−144
−143	3.556	7.240	0.01724	5.305	5.305	−216.50	1214.39	997.89	−0.4727	3.8346	3.3619	−143
−142	3.967	8.076	0.01724	4.770	4.770	−216.16	1214.49	998.33	−0.4716	3.8229	3.3513	−142
−141	4.423	9.005	0.01724	4.292	4.292	−215.83	1214.60	998.77	−0.4706	3.8112	3.3406	−141
−140	4.928	10.03	0.01724	3.864	3.864	−215.49	1214.70	999.21	−0.4695	3.7996	3.3301	−140
−139	5.487	11.17	0.01724	3.481	3.481	−215.16	1214.82	999.66	−0.4685	3.7881	3.3196	−139
−138	6.106	12.43	0.01724	3.138	3.138	−214.82	1214.92	1000.10	−0.4674	3.7766	3.3092	−138
−137	6.790	13.82	0.01724	2.831	2.831	−214.48	1215.02	1000.54	−0.4664	3.7653	3.2989	−137
−136	7.546	15.36	0.01725	2.555	2.555	−214.14	1215.12	1000.98	−0.4653	3.7540	3.2887	−136
−135	8.380	17.06	0.01725	2.308	2.308	−213.80	1215.22	1001.42	−0.4643	3.7428	3.2785	−135
−134	9.301	18.94	0.01725	2.086	2.086	−213.46	1215.32	1001.86	−0.4632	3.7315	3.2683	−134
−133	10.32	21.01	0.01725	1.886	1.886	−213.11	1215.42	1002.31	−0.4622	3.7205	3.2583	−133
−132	11.44	23.28	0.01725	1.707	1.707	−212.77	1215.52	1002.75	−0.4611	3.7093	3.2482	−132
−131	12.67	25.79	0.01725	1.545	1.545	−212.43	1215.62	1003.19	−0.4601	3.6984	3.2383	−131
−130	14.03	28.56	0.01725	1.400	1.400	−212.08	1215.71	1003.63	−0.4590	3.6874	3.2284	−130
−129	15.52	31.60	0.01725	1.269	1.269	−211.73	1215.80	1004.07	−0.4580	3.6766	3.2186	−129

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^6$		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-7}$	Sat. Vapor $v_g \times 10^{-7}$	Sat. Solid h_i	Evap h_{ig}	Sat Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat Vapor s_g	
−128	1.716	3.494	0.01726	11.51	11.51	−211.39	1215.91	1004.52	−0.4569	3.6658	3.2089	−128
−127	1.897	3.862	0.01726	10.45	10.45	−211.04	1216.00	1004.96	−0.4559	3.6551	3.1992	−127
−126	2.095	4.265	0.01726	9.489	9.489	−210.69	1216.09	1005.40	−0.4548	3.6444	3.1896	−126
−125	2.312	4.708	0.01726	8.622	8.622	−210.34	1216.18	1005.84	−0.4538	3.6338	3.1800	−125
−124	2.551	5.194	0.01726	7.839	7.839	−209.99	1216.27	1006.28	−0.4527	3.6232	3.1705	−124
−123	2.812	5.726	0.01726	7.131	7.131	−209.64	1216.37	1006.73	−0.4517	3.6127	3.1610	−123
−122	3.099	6.310	0.01726	6.491	6.491	−209.28	1216.45	1007.17	−0.4506	3.6022	3.1516	−122
−121	3.413	6.949	0.01726	5.911	5.911	−208.93	1216.54	1007.61	−0.4496	3.5919	3.1423	−121
−120	3.757	7.649	0.01726	5.386	5.386	−208.58	1216.63	1008.05	−0.4485	3.5815	3.1330	−120
−119	4.133	8.414	0.01726	4.911	4.911	−208.22	1216.71	1008.49	−0.4475	3.5713	3.1238	−119
−118	4.544	9.251	0.01727	4.480	4.480	−207.86	1216.80	1008.94	−0.4464	3.5611	3.1147	−118
−117	4.993	10.17	0.01727	4.088	4.088	−207.51	1216.89	1009.38	−0.4454	3.5510	3.1056	−117
−116	5.484	11.16	0.01727	3.733	3.733	−207.15	1216.97	1009.82	−0.4444	3.5409	3.0965	−116
−115	6.019	12.26	0.01727	3.411	3.411	−206.79	1217.05	1010.26	−0.4433	3.5308	3.0875	−115
−114	6.604	13.44	0.01727	3.118	3.118	−206.43	1217.13	1010.70	−0.4423	3.5209	3.0786	−114
−113	7.241	14.74	0.01727	2.852	2.852	−206.07	1217.21	1011.14	−0.4412	3.5109	3.0697	−113
−112	7.936	16.16	0.01727	2.610	2.610	−205.70	1217.29	1011.59	−0.4402	3.5011	3.0609	−112
−111	8.693	17.70	0.01727	2.389	2.389	−205.34	1217.37	1012.03	−0.4391	3.4912	3.0521	−111
−110	9.517	19.38	0.01728	2.189	2.189	−204.98	1217.45	1012.47	−0.4381	3.4815	3.0434	−110
−109	10.41	21.20	0.01728	2.006	2.006	−204.61	1217.52	1012.91	−0.4370	3.4718	3.0348	−109
−108	11.39	23.19	0.01728	1.839	1.839	−204.24	1217.60	1013.36	−0.4360	3.4621	3.0261	−108
−107	12.45	25.35	0.01728	1.687	1.687	−203.88	1217.68	1013.80	−0.4350	3.4526	3.0176	−107
−106	13.60	27.70	0.01728	1.549	1.549	−203.51	1217.75	1014.24	−0.4339	3.4430	3.0091	−106
−105	14.86	30.25	0.01728	1.422	1.422	−203.14	1217.82	1014.68	−0.4329	3.4335	3.0006	−105
−104	16.22	33.01	0.01728	1.307	1.307	−202.77	1217.89	1015.12	−0.4318	3.4240	2.9922	−104
−103	17.69	36.02	0.01728	1.201	1.201	−202.40	1217.96	1015.56	−0.4308	3.4146	2.9838	−103
−102	19.29	39.28	0.01728	1.104	1.104	−202.03	1218.04	1016.01	−0.4298	3.4053	2.9755	−102
−101	21.03	42.81	0.01729	1.016	1.016	−201.65	1218.10	1016.45	−0.4287	3.3960	2.9673	−101
−100	22.91	46.64	0.01729	0.9352	0.9352	−201.28	1218.17	1016.89	−0.4277	3.3868	2.9591	−100
−99	24.94	50.79	0.01729	0.8613	0.8613	−200.90	1218.23	1017.33	−0.4266	3.3775	2.9509	−99
−98	27.15	55.28	0.01729	0.7936	0.7936	−200.53	1218.30	1017.77	−0.4256	3.3684	2.9428	−98
−97	29.54	60.14	0.01729	0.7814	0.7814	−200.15	1218.37	1018.22	−0.4246	3.3593	2.9347	−97

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^5$		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-6}$	Sat. Vapor $v_g \times 10^{-6}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−96	3.212	6.539	0 01729	6.745	6.745	−199.78	1218.44	1018.66	−0 4235	3.3502	2 9267	−96
−95	3.491	7.108	0.01729	6.223	6 223	−199.40	1218.50	1019.10	−0.4225	3.3412	2.9187	−95
−94	3.793	7.722	0.01729	5.743	5.743	−199.02	1218.56	1019.54	−0.4214	3.3322	2.9108	−94
−93	4.118	8.385	0.01730	5.303	5.303	−198.64	1218.62	1019.98	−0.4204	3.3233	2 9029	−93
−92	4.470	9.102	0 01730	4.899	4.899	−198.25	1218.68	1020.43	−0.4194	3.3145	2.8951	−92
−91	4.850	9.876	0 01730	4.528	4.528	−197.87	1218.74	1020.87	−0.4183	3.3056	2.8873	−91
−90	5.260	10.71	0.01730	4.186	4.186	−197.49	1218.80	1021.31	−0.4173	3.2969	2.8796	−90
−89	5.702	11.61	0.01730	3.872	3.872	−197.10	1218.85	1021.75	−0 4162	3.2880	2.8718	−89
−88	6.179	12.58	0.01730	3.583	3.583	−196.72	1218.92	1022.20	−0.4152	3.2794	2.8642	−88
−87	6 692	13.63	0.01730	3 317	3.317	−196.33	1218.97	1022 64	−0.4142	3.2708	2.8566	−87
−86	7.245	14.75	0.01730	3.072	3.072	−195.94	1219 02	1023 08	−0.4131	3.2621	2.8490	−86
−85	7.841	15.96	0.01730	2.846	2.846	−195.56	1219.08	1023.52	−0.4121	3.2536	2.8415	−85
−84	8.482	17.27	0 01731	2.638	2 638	−195.16	1219 12	1023.96	−0 4110	3.2450	2.8340	−84
−83	9.171	18.67	0.01731	2.446	2.446	−194.78	1219.18	1024.40	−0.4100	3.2366	2.8266	−83
−82	9.913	20 18	0 01731	2.270	2.270	−194.38	1219.23	1024.85	−0.4090	3.2282	2.8192	−82
−81	10.71	21.81	0.01731	2.106	2.106	−193.99	1219.28	1025.29	−0.4079	3 2197	2 8118	−81
−80	11.57	23.55	0.01731	1.955	1.955	−193.60	1219.33	1025.73	−0.4069	3.2114	2.8045	−80
−79	12 49	25.42	0.01731	1.816	1.816	−193 20	1219.37	1026.17	−0 4058	3 2030	2 7972	−79
−78	13.47	27 43	0 01731	1.687	1 687	−192.81	1219.43	1026 62	−0.4048	3.1948	2.7900	−78
−77	14.53	29 59	0.01732	1 568	1.568	−192.41	1219.47	1027.06	−0.4038	3.1866	2 7828	−77
−76	15.67	31.91	0.01732	1.458	1.458	−192.01	1219.51	1027.50	−0.4028	3.1784	2.7756	−76
−75	16 89	34.39	0.01732	1.356	1.356	−191.62	1219.56	1027 94	−0.4017	3.1702	2 7685	−75
−74	18.20	37 05	0.01732	1.262	1.262	−191 22	1219.60	1028.38	−0 4007	3.1622	2.7615	−74
−73	19.60	39.91	0.01732	1.175	1.175	−190.82	1219.65	1028.83	−0.3996	3.1540	2 7544	−73
−72	21.10	42.96	0.01732	1 094	1 094	−190 41	1219.68	1029.27	−0 3986	3.1460	2 7474	−72
−71	22.71	46 24	0 01732	1.020	1 020	−190.01	1219 72	1029.71	−0.3976	3 1381	2.7405	−71
−70	24.43	49.74	0.01732	0 9501	0.9501	−189 61	1219.76	1030.15	−0.3965	3.1301	2.7336	−70
−69	26.27	53.49	0.01732	0.8858	0 8858	−189.20	1219.80	1030 60	−0.3955	3.1222	2.7267	−69
−68	28.24	57.51	0.01733	0.8261	0.8261	−188.80	1219.84	1031 04	−0.3945	3 1143	2.7198	−68
−67	30.35	61 80	0.01733	0.7707	0 7707	−188.39	1219 87	1031 48	−0.3934	3.1064	2.7130	−67
−66	32.60	66 38	0.01733	0.7193	0.7193	−187.98	1219 90	1031.92	−0 3924	3 0987	2.7063	−66
−65	35.01	71.28	0.01733	0.6715	0 6715	−187 58	1219.94	1032.36	−0.3914	3.0910	2.6996	−65

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp t(F)	Absolute Pressure $p_s \times 10^3$		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr Temp. t(F)
	Lb/Sq In.	In. Hg	Sat Solid v_i	Evap. $v_{ig} \times 10^{-5}$	Sat. Vapor $v_g \times 10^{-5}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat Vapor s_g	
−64	0.3758	0.7652	0.01733	6.272	6.272	−187.17	1219.98	1032.81	−0.3903	3.0832	2.6929	−64
−63	0.4033	0.8211	0.01733	5.859	5.859	−186.76	1220.01	1033.25	−0.3893	3.0755	2.6862	−63
−62	0.4326	0.8808	0.01734	5.476	5.476	−186.35	1220.04	1033.69	−0.3882	3.0678	2.6796	−62
−61	0.4639	0.9444	0.01734	5.120	5.120	−185.93	1220.06	1034.13	−0.3872	3.0602	2.6730	−61
−60	0.4972	1.012	0.01734	4.788	4.788	−185.52	1220.10	1034.58	−0.3862	3.0526	2.6664	−60
−59	0.5328	1.085	0.01734	4.479	4.479	−185.11	1220.13	1035.02	−0.3851	3.0450	2.6599	−59
−58	0.5708	1.162	0.01734	4.192	4.192	−184.69	1220.15	1035.46	−0.3841	3.0376	2.6535	−58
−57	0.6112	1.244	0.01734	3.925	3.925	−184.28	1220.18	1035.90	−0.3831	3.0301	2.6470	−57
−56	0.6543	1.332	0.01734	3.675	3.675	−183.86	1220.20	1036.34	−0.3820	3.0226	2.6406	−56
−55	0.7001	1.426	0.01734	3.443	3.443	−183.44	1220.23	1036.79	−0.3810	3.0152	2.6342	−55
−54	0.7489	1.525	0.01735	3.226	3.226	−183.02	1220.25	1037.23	−0.3800	3.0079	2.6279	−54
−53	0.8009	1.631	0.01735	3.024	3.024	−182.60	1220.27	1037.67	−0.3789	3.0005	2.6216	−53
−52	0.8562	1.743	0.01735	2.836	2.836	−182.18	1220.29	1038.11	−0.3779	2.9932	2.6153	−52
−51	0.9151	1.863	0.01735	2.660	2.660	−181.76	1220.31	1038.55	−0.3769	2.9860	2.6091	−51
−50	0.9776	1.990	0.01735	2.496	2.496	−181.34	1220.34	1039.00	−0.3758	2.9786	2.6028	−50
−49	1.044	2.126	0.01735	2.343	2.343	−180.92	1220.36	1039.44	−0.3748	2.9715	2.5967	−49
−48	1.115	2.270	0.01736	2.200	2.200	−180.49	1220.37	1039.88	−0.3738	2.9643	2.5905	−48
−47	1.190	2.422	0.01736	2.066	2.066	−180.06	1220.38	1040.32	−0.3727	2.9571	2.5844	−47
−46	1.270	2.585	0.01736	1.941	1.941	−179.64	1220.40	1040.76	−0.3717	2.9501	2.5784	−46
−45	1.354	2.757	0.01736	1.824	1.824	−179.21	1220.42	1041.21	−0.3707	2.9430	2.5723	−45
−44	1.444	2.940	0.01736	1.715	1.715	−178.78	1220.43	1041.65	−0.3696	2.9359	2.5663	−44
−43	1.539	3.134	0.01736	1.612	1.612	−178.35	1220.44	1042.09	−0.3686	2.9289	2.5603	−43
−42	1.641	3.340	0.01736	1.516	1.516	−177.92	1220.45	1042.53	−0.3676	2.9219	2.5543	−42
−41	1.748	3.559	0.01736	1.427	1.427	−177.49	1220.47	1042.98	−0.3665	2.0149	2.5484	−41
−40	1.861	3.790	0.01737	1.343	1.343	−177.06	1220.48	1043.42	−0.3655	2.9080	2.5425	−40
−39	1.982	4.035	0.01737	1.264	1.264	−176.63	1220.49	1043.86	−0.3645	2.9012	2.5367	−39
−38	2.110	4.295	0.01737	1.191	1.191	−176.19	1220.49	1044.30	−0.3634	2.8942	2.5308	−38
−37	2.245	4.570	0.01737	1.122	1.122	−175.76	1220.50	1044.74	−0.3624	2.8874	2.5250	−37
−36	2.388	4.862	0.01737	1.057	1.057	−175.32	1220.51	1045.19	−0.3614	2.8807	2.5193	−36
−35	2.540	5.170	0.01737	0.9961	0.9961	−174.88	1220.51	1045.63	−0.3604	2.8739	2.5135	−35
−34	2.700	5.497	0.01737	0.9391	0.9391	−174.45	1220.52	1046.07	−0.3593	2.8671	2.5078	−34
−33	2.870	5.843	0.01738	0.8857	0.8857	−174.01	1220.52	1046.51	−0.3583	2.8604	2.5021	−33

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^2$		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap $v_{ig} \times 10^{-4}$	Sat Vapor $v_g \times 10^{-4}$	Sat. Solid h_i	Evap h_{ig}	Sat Vapor h_g	Sat Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−32	0.3049	0.6208	0.01738	8.355	8.355	−173.57	1220.52	1046.95	−0.3573	2.8538	2.4965	−32
−31	0.3239	0.6595	0.01738	7.883	7.883	−173.13	1220.53	1047.40	−0.3562	2.8470	2.4908	−31
−30	0.3440	0.7003	0.01738	7.441	7.441	−172.68	1220.52	1047.84	−0.3552	2.8405	2.4853	−30
−29	0.3652	0.7435	0.01738	7.025	7.025	−172.24	1220.52	1048.28	−0.3542	2.8339	2.4797	−29
−28	0.3876	0.7891	0.01738	6.634	6.634	−171.80	1220.52	1048.72	−0.3532	2.8274	2.4742	−28
−27	0.4113	0.8373	0.01738	6.267	6.267	−171.35	1220.51	1049.16	−0.3521	2.8207	2.4686	−27
−26	0.4363	0.8882	0.01738	5.921	5.921	−170.91	1220.51	1049.60	−0.3511	2.8143	2.4632	−26
−25	0.4627	0.9420	0.01739	5.596	5.596	−170.46	1220.51	1050.05	−0.3501	2.8078	2.4577	−25
−24	0.4905	0.9987	0.01739	5.290	5.290	−170.01	1220.50	1050.49	−0.3490	2.8013	2.4523	−24
−23	0.5199	1.059	0.01739	5.003	5.003	−169.56	1220.49	1050.93	−0.3480	2.7940	2.4460	−23
−22	0.5509	1.122	0.01739	4.732	4.732	−169.12	1220.49	1051.37	−0.3470	2.7885	2.4415	−22
−21	0.5836	1.188	0.01739	4.477	4.477	−168.66	1220.48	1051.82	−0.3460	2.7822	2.4362	−21
−20	0.6181	1.259	0.01739	4.237	4.237	−168.21	1220.47	1052.26	−0.3449	2.7757	2.4308	−20
−19	0.6545	1.333	0.01739	4.011	4.011	−167.76	1220.46	1052.70	−0.3439	2.7695	2.4256	−19
−18	0.6928	1.410	0.01740	3.797	3.797	−167.31	1220.45	1053.14	−0.3429	2.7632	2.4203	−18
−17	0.7332	1.493	0.01740	3.596	3.596	−166.85	1220.43	1053.58	−0.3418	2.7568	2.4150	−17
−16	0.7757	1.579	0.01740	3.407	3.407	−166.40	1220.42	1054.02	−0.3408	2.7506	2.4098	−16
−15	0.8204	1.670	0.01740	3.228	3.228	−165.94	1220.41	1054.47	−0.3398	2.7444	2.4046	−15
−14	0.8676	1.766	0.01740	3.060	3.060	−165.48	1220.39	1054.91	−0.3388	2.7383	2.3995	−14
−13	0.9172	1.867	0.01740	2.901	2.901	−165.03	1220.38	1055.35	−0.3377	2.7320	2.3943	−13
−12	0.9694	1.974	0.01740	2.750	2.750	−164.57	1220.36	1055.79	−0.3367	2.7259	2.3892	−12
−11	1.024	2.086	0.01740	2.609	2.609	−164.11	1220.34	1056.23	−0.3357	2.7198	2.3841	−11
−10	1.082	2.203	0.01741	2.475	2.475	−163.65	1220.32	1056.67	−0.3347	2.7138	2.3791	−10
−9	1.143	2.327	0.01741	2.349	2.349	−163.18	1220.30	1057.12	−0.3336	2.7076	2.3740	−9
−8	1.207	2.457	0.01741	2.229	2.229	−162.72	1220.28	1057.56	−0.3326	2.7016	2.3690	−8
−7	1.274	2.594	0.01741	2.116	2.116	−162.26	1220.26	1058.00	−0.3316	2.6956	2.3640	−7
−6	1.344	2.737	0.01741	2.010	2.010	−161.79	1220.23	1058.44	−0.3306	2.6896	2.3590	−6
−5	1.419	2.888	0.01741	1.909	1.909	−161.33	1220.21	1058.88	−0.3295	2.6836	2.3541	−5
−4	1.496	3.047	0.01742	1.814	1.814	−160.86	1220.18	1059.32	−0.3285	2.6777	2.3492	−4
−3	1.578	3.213	0.01742	1.723	1.723	−160.39	1220.15	1059.76	−0.3275	2.6718	2.3443	−3
−2	1.664	3.388	0.01742	1.638	1.638	−159.92	1220.13	1060.21	−0.3264	2.6658	2.3394	−2
−1	1.754	3.572	0.01742	1.557	1.557	−159.45	1220.10	1060.65	−0.3254	2.6600	2.3346	−1

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr Temp. t(F)
	Lb/Sq In	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-3}$	Sat. Vapor $v_g \times 10^{-3}$	Sat Solid h_i	Evap h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
0	0.01849	0.03764	0.01742	14.81	14.81	−158.98	1220.07	1061.09	−0.3244	2.6541	2.3297	0
1	0.01948	0.03966	0.01742	14.08	14.08	−158.51	1220.04	1061.53	−0.3234	2.6483	2.3249	1
2	0.02052	0.04178	0.01742	13.40	13.40	−158.04	1220.01	1061.97	−0.3224	2.6425	2.3201	2
3	0.02616	0.04400	0.01743	12.75	12.75	−157.56	1219.97	1062.41	−0.3213	2.6367	2.3154	3
4	0.02276	0.04633	0.01743	12.14	12.14	−157.09	1219.94	1062.85	−0.3203	2.6309	2.3106	4
5	0.02396	0.04878	0.01743	11.55	11.55	−156.61	1219.90	1063.29	−0.3193	2.6252	2.3059	5
6	0.02521	0.05134	0.01743	11.00	11.00	−156.14	1219.88	1063.74	−0.3182	2.6194	2.3012	6
7	0.02653	0.05402	0.01743	10.48	10.48	−155.66	1219.84	1064.18	−0.3172	2.6138	2.2966	7
8	0.02791	0.05683	0.01743	9.979	9.979	−155.18	1219.80	1064.62	−0.3162	2.6081	2.2919	8
9	0.02936	0.05977	0.01744	9.507	9.507	−154.70	1219.76	1065.06	−0.3152	2.6025	2.2873	9
10	0.03087	0.06286	0.01744	9.060	9.060	−154.22	1219.72	1065.50	−0.3142	2.5969	2.2827	10
11	0.03246	0.06608	0.01744	8.636	8.636	−153.74	1219.68	1065.94	−0.3131	2.5912	2.2781	11
12	0.03412	0.06946	0.01744	8.234	8.234	−153.26	1219.64	1066.38	−0.3121	2.5857	2.2736	12
13	0.03585	0.07300	0.01744	7.851	7.851	−152.77	1219.59	1066.82	−0.3111	2.5801	2.2690	13
14	0.03767	0.07669	0.01744	7.489	7.489	−152.29	1219.55	1067.26	−0.3101	2.5746	2.2645	14
15	0.03957	0.08056	0.01744	7.144	7.144	−151.80	1219.50	1067.70	−0.3090	2.5690	2.2600	15
16	0.04156	0.08461	0.01745	6.817	6.817	−151.32	1219.46	1068.14	−0.3080	2.5635	2.2555	16
17	0.04363	0.08884	0.01745	6.505	6.505	−150.83	1219.41	1068.58	−0.3070	2.5581	2.2511	17
18	0.04581	0.09326	0.01745	6.210	6.210	−150.34	1219.36	1069.02	−0.3060	2.5526	2.2466	18
19	0.04808	0.09789	0.01745	5.929	5.929	−149.85	1219.31	1069.46	−0.3049	2.5471	2.2422	19
20	0.05045	0.1027	0.01745	5.662	5.662	−149.36	1219.26	1069.90	−0.3039	2.5417	2.2378	20
21	0.05293	0.1078	0.01745	5.408	5.408	−148.87	1219.21	1070.34	−0.3029	2.5364	2.2335	21
22	0.05552	0.1130	0.01746	5.166	5.166	−148.38	1219.16	1070.78	−0.3019	2.5310	2.2291	22
23	0.05823	0.1186	0.01746	4.936	4.936	−147.88	1219.10	1071.22	−0.3008	2.5256	2.2248	23
24	0.06105	0.1243	0.01746	4.717	4.717	−147.39	1219.05	1071.66	−0.2998	2.5203	2.2205	24
25	0.06400	0.1303	0.01746	4.509	4.509	−146.89	1218.98	1072.09	−0.2988	2.5150	2.2162	25
26	0.06708	0.1366	0.01746	4.311	4.311	−146.40	1218.93	1072.53	−0.2978	2.5097	2.2119	26
27	0.07030	0.1431	0.01746	4.122	4.122	−145.90	1218.87	1072.97	−0.2968	2.5045	2.2077	27
28	0.07365	0.1500	0.01746	3.943	3.943	−145.40	1218.81	1073.41	−0.2957	2.4991	2.2034	28
29	0.07715	0.1571	0.01747	3.771	3.771	−144.90	1218.75	1073.85	−0.2947	2.4939	2.1992	29
30	0.08080	0.1645	0.01747	3.608	3.608	−144.40	1218.69	1074.29	−0.2937	2.4887	2.1950	30
31	0.08461	0.1723	0.01747	3.453	3.453	−143.90	1218.63	1074.73	−0.2927	2.4835	2.1908	31
32	0.08858	0.1803	0.01747	3.305	3.305	−143.40	1218.56	1075.16	−0.2916	2.4783	2.1867	32

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat Liquid v_f	Evap v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
*32	0.088586	0.18036	0.01602	3304.6	3304.6	0.00	1075.16	1075.16	0.00000	2.1867	2.1867	32*
33	0.092227	0.18778	0.01602	3180.5	3180.5	1.01	1074.59	1075.60	0.00205	2.1811	2.1831	33
34	0.095999	0.19546	0.01602	3061.7	3061.7	2.01	1074.03	1076.04	0.00409	2.1755	2.1796	34
35	0.099908	0.20342	0.01602	2947.8	2947.8	3.02	1073.46	1076.48	0.00612	2.1700	2.1761	35
36	0.10396	0.21166	0.01602	2838.7	2838.7	4.02	1072.90	1076.92	0.00815	2.1644	2.1726	36
37	0.10815	0.22020	0.01602	2734.1	2734.1	5.03	1072.33	1077.36	0.01018	2.1589	2.1691	37
38	0.11249	0.22904	0.01602	2633.8	2633.8	6.03	1071.77	1077.80	0.01220	2.1535	2.1657	38
39	0.11699	0.23819	0.01602	2537.6	2537.6	7.04	1071.20	1078.24	0.01422	2.1480	2.1622	39
40	0.12164	0.24767	0.01602	2445.4	2445.4	8.04	1070.64	1078.68	0.01623	2.1426	2.1588	40
41	0.12646	0.25748	0.01602	2356.9	2356.9	9.05	1070.06	1079.11	0.01824	2.1372	2.1554	41
42	0.13145	0.26763	0.01602	2272.0	2272.0	10.05	1069.50	1079.55	0.02024	2.1318	2.1520	42
43	0.13660	0.27813	0.01602	2190.5	2190.5	11.05	1068.94	1079.99	0.02224	2.1265	2.1487	43
44	0.14194	0.28899	0.01602	2112.3	2112.3	12.06	1068.37	1080.43	0.02423	2.1211	2.1453	44
45	0.14746	0.30023	0.01602	2037.3	2037.3	13.06	1067.81	1080.87	0.02622	2.1158	2.1420	45
46	0.15317	0.31185	0.01602	1965.2	1965.2	14.06	1067.24	1081.30	0.02820	2.1105	2.1387	46
47	0.15907	0.32387	0.01602	1896.0	1896.0	15.06	1066.68	1081.74	0.03018	2.1052	2.1354	47
48	0.16517	0.33629	0.01602	1829.5	1829.5	16.07	1066.11	1082.18	0.03216	2.0999	2.1321	48
49	0.17148	0.34913	0.01602	1765.7	1765.7	17.07	1065.55	1082.62	0.03413	2.0947	2.1288	49
50	0.17799	0.36240	0.01602	1704.3	1704.3	18.07	1064.99	1083.06	0.03610	2.0895	2.1256	50
51	0.18473	0.37611	0.01602	1645.4	1645.4	19.07	1064.42	1083.49	0.03806	2.0842	2.1223	51
52	0.19169	0.39028	0.01602	1588.7	1588.7	20.07	1063.86	1083.93	0.04002	2.0791	2.1191	52
53	0.19888	0.40492	0.01603	1534.3	1534.3	21.07	1063.30	1084.37	0.04197	2.0739	2.1159	53
54	0.20630	0.42003	0.01603	1481.9	1481.9	22.08	1062.72	1084.80	0.04392	2.0688	2.1127	54
55	0.21397	0.43564	0.01603	1431.5	1431.5	23.08	1062.16	1085.24	0.04587	2.0637	2.1096	55
56	0.22188	0.45176	0.01603	1383.1	1383.1	24.08	1061.60	1085.68	0.04781	2.0586	2.1064	56
57	0.23006	0.46840	0.01603	1336.5	1336.5	25.08	1061.04	1086.12	0.04975	2.0535	2.1033	57
58	0.23849	0.48558	0.01603	1291.7	1291.7	26.08	1060.47	1086.55	0.05168	2.0485	2.1002	58
59	0.24720	0.50330	0.01603	1248.6	1248.6	27.08	1059.91	1086.99	0.05361	2.0434	2.0970	59
60	0.25618	0.52160	0.01603	1207.1	1207.1	28.08	1059.34	1087.42	0.05553	2.0385	2.0940	60
61	0.26545	0.54047	0.01604	1167.2	1167.2	29.08	1058.78	1087.86	0.05746	2.0334	2.0909	61

[a]Compiled by John A. Goff and S. Gratch.
*Extrapolated to represent metastable equilibrium with undercooled liquid.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In	In. Hg	Sat Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
62	0.27502	0.55994	0.01604	1128.7	1128.7	30.08	1058.22	1088.30	0.05937	2.0284	2.0878	62
63	0.28488	0.58002	0.01604	1091.7	1091.7	31.08	1057.65	1088.73	0.06129	2.0235	2.0848	63
64	0.29505	0.60073	0.01604	1056.1	1056.1	32.08	1057.09	1089.17	0.06320	2.0186	2.0818	64
65	0.30554	0.62209	0.01604	1021.7	1021.7	33.08	1056.52	1089.60	0.06510	2.0136	2.0787	65
66	0.31636	0.64411	0.01604	988.63	988.65	34.07	1055.97	1090.04	0.06700	2.0087	2.0757	66
67	0.32750	0.66681	0.01605	956.76	956.78	35.07	1055.40	1090.47	0.06890	2.0039	2.0728	67
68	0.33900	0.69021	0.01605	926.06	926.08	36.07	1054.84	1090.91	0.07080	1.9990	2.0698	68
69	0.35084	0.71432	0.01605	896.47	896.49	37.07	1054.27	1091.34	0.07269	1.9941	2.0668	69
70	0.36304	0.73916	0.01605	867.95	867.97	38.07	1053.71	1091.78	0.07458	1.9893	2.0639	70
71	0.37561	0.76476	0.01605	840.45	840.47	39.07	1053.14	1092.21	0.07646	1.9845	2.0610	71
72	0.38856	0.79113	0.01606	813.95	813.97	40.07	1052.58	1092.65	0.07834	1.9797	2.0580	72
73	0.40190	0.81829	0.01606	788.38	788.40	41.07	1052.01	1093.08	0.08022	1.9749	2.0551	73
74	0.41564	0.84626	0.01606	763.73	763.75	42.06	1051.46	1093.52	0.08209	1.9701	2.0522	74
75	0.42979	0.87506	0.01606	739.95	739.97	43.06	1050.89	1093.95	0.08396	1.9654	2.0494	75
76	0.44435	0.90472	0.01606	717.01	717.03	44.06	1050.32	1094.38	0.08582	1.9607	2.0465	76
77	0.45935	0.93524	0.01607	694.88	694.90	45.06	1049.76	1094.82	0.08769	1.9560	2.0437	77
78	0.47478	0.96666	0.01607	673.52	673.54	46.06	1049.19	1095.25	0.08954	1.9513	2.0408	78
79	0.49066	0.99900	0.01607	652.91	652.93	47.06	1048.62	1095.68	0.09140	1.9466	2.0380	79
80	0.50701	1.0323	0.01607	633.01	633.03	48.05	1048.07	1096.12	0.09325	1.9419	2.0352	80
81	0.52382	1.0665	0.01608	613.80	613.82	49.05	1047.50	1096.55	0.09510	1.9373	2.0324	81
82	0.54112	1.1017	0.01608	595.25	595.27	50.05	1046.93	1096.98	0.09694	1.9328	2.0297	82
83	0.55892	1.1380	0.01608	577.34	577.36	51.05	1046.37	1097.42	0.09878	1.9281	2.0269	83
84	0.57722	1.1752	0.01608	560.04	560.06	52.05	1045.80	1097.85	0.10062	1.9236	2.0242	84
85	0.59604	1.2136	0.01609	543.33	543.35	53.05	1045.23	1098.28	0.10246	1.9189	2.0214	85
86	0.61540	1.2530	0.01609	527.19	527.21	54.04	1044.67	1098.71	0.10429	1.9144	2.0187	86
87	0.63530	1.2935	0.01609	511.60	511.62	55.04	1044.10	1099.14	0.10611	1.9099	2.0160	87
88	0.65575	1.3351	0.01610	496.52	496.54	56.04	1043.54	1099.58	0.10794	1.9054	2.0133	88
89	0.67678	1.3779	0.01610	481.96	481.98	57.04	1042.97	1100.01	0.10976	1.9008	2.0106	89
90	0.69838	1.4219	0.01610	467.88	467.90	58.04	1042.40	1100.44	0.11158	1.8963	2.0079	90
91	0.72059	1.4671	0.01610	454.26	454.28	59.03	1041.84	1100.87	0.11339	1.8919	2.0053	91

[a]Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap v_{fg}	Sat Vapor v_g	Sat Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
92	0.74340	1.5136	0.01611	441.10	441.12	60.03	1041.27	1101.30	0.11520	1.8874	2.0026	92
93	0.76684	1.5613	0.01611	428.38	428.40	61.03	1040.70	1101.73	0.11701	1.8830	2.0000	93
94	0.79091	1.6103	0.01611	416.07	416.09	62.03	1040.13	1102.16	0.11881	1.8786	1.9974	94
95	0.81564	1.6607	0.01612	404.17	404.19	63.03	1039.56	1102.59	0.12061	1.8741	1.9947	95
96	0.84103	1.7124	0.01612	392.65	392.67	64.02	1039.00	1103.02	0.12241	1.8698	1.9922	96
97	0.86711	1.7655	0.01612	381.51	381.53	65.02	1038.43	1103.45	0.12420	1.8654	1.9896	97
98	0.89388	1.8200	0.01612	370.73	370.75	66.02	1037.86	1103.88	0.12600	1.8610	1.9870	98
99	0.92137	1.8759	0.01613	360.30	360.32	67.02	1037.29	1104.31	0.12778	1.8566	1.9844	99
100	0.94959	1.9334	0.01613	350.20	350.22	68.02	1036.72	1104.74	0.12957	1.8523	1.9819	100
101	0.97854	1.9923	0.01614	340.42	340.44	69.01	1036.16	1105.17	0.13135	1.8480	1.9793	101
102	1.0083	2.0529	0.01614	330.96	330.98	70.01	1035.58	1105.59	0.13313	1.8437	1.9768	102
103	1.0388	2.1149	0.01614	321.80	321.82	71.01	1035.01	1106.02	0.13490	1.8394	1.9743	103
104	1.0700	2.1786	0.01614	312.93	312.95	72.01	1034.44	1106.45	0.13667	1.8351	1.9718	104
105	1.1021	2.2440	0.01615	304.34	304.36	73.01	1033.87	1106.88	0.13844	1.8309	1.9693	105
106	1.1351	2.3110	0.01615	296.02	296.04	74.01	1033.29	1107.30	0.14021	1.8266	1.9668	106
107	1.1688	2.3798	0.01616	287.96	287.98	75.00	1032.73	1107.73	0.14197	1.8224	1.9644	107
108	1.2035	2.4503	0.01616	280.14	280.16	76.00	1032.16	1108.16	0.14373	1.8182	1.9619	108
109	1.2390	2.5226	0.01616	272.58	272.60	77.00	1031.58	1108.58	0.14549	1.8140	1.9595	109
110	1.2754	2.5968	0.01617	265.24	265.26	78.00	1031.01	1109.01	0.14724	1.8098	1.9570	110
111	1.3128	2.6728	0.01617	258.14	258.16	79.00	1030.44	1109.44	0.14899	1.8056	1.9546	111
112	1.3510	2.7507	0.01617	251.25	251.27	80.00	1029.86	1109.86	0.15074	1.8015	1.9522	112
113	1.3902	2.8306	0.01618	244.57	244.59	80.99	1029.30	1110.29	0.15248	1.7973	1.9498	113
114	1.4305	2.9125	0.01618	238.10	238.12	81.99	1028.72	1110.71	0.15423	1.7932	1.9474	114
115	1.4717	2.9963	0.01618	231.82	231.84	82.99	1028.15	1111.14	0.15596	1.7890	1.9450	115
116	1.5139	3.0823	0.01619	225.73	225.75	83.99	1027.57	1111.56	0.15770	1.7849	1.9426	116
117	1.5571	3.1703	0.01619	219.83	219.85	84.99	1026.99	1111.98	0.15943	1.7809	1.9403	117
118	1.6014	3.2606	0.01620	214.10	214.12	85.99	1026.42	1112.41	0.16116	1.7767	1.9379	118
119	1.6468	3.3530	0.01620	208.54	208.56	86.98	1025.85	1112.83	0.16289	1.7727	1.9356	119
120	1.6933	3.4477	0.01620	203.16	203.18	87.98	1025.28	1113.26	0.16461	1.7687	1.9333	120
121	1.7409	3.5446	0.01621	197.93	197.95	88.98	1024.70	1113.68	0.16634	1.7647	1.9310	121

[a]Compiled by John A. Goff and S Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat Liquid v_f	Evap v_{fg}	Sat. Vapor v_g	Sat Liquid h_f	Evap h_{fg}	Sat Vapor h_g	Sat. Liquid s_f	Evap s_{fg}	Sat. Vapor s_g	
122	1.7897	3.6439	0.01621	192.85	192.87	89.98	1024.12	1114.10	0.16805	1.7606	1.9286	122
123	1.8396	3.7455	0.01622	187.93	187.95	90.98	1023.54	1114.52	0.16977	1.7566	1.9264	123
124	1.8907	3.8496	0.01622	183.15	183.17	91.98	1022.96	1114.94	0.17148	1.7526	1.9241	124
125	1.9430	3.9561	0.01622	178.51	178.53	92.98	1022.39	1115.37	0.17319	1.7486	1.9218	125
126	1.9966	4.0651	0.01623	174.00	174.02	93.98	1021.81	1115.79	0.17490	1.7446	1.9195	126
127	2.0514	4.1768	0.01623	169.63	169.65	94.97	1021.24	1116.21	0.17660	1.7407	1.9173	127
128	2.1075	4.2910	0.01624	165.38	165.40	95.97	1020.66	1116.63	0.17830	1.7367	1.9150	128
129	2.1649	4.4078	0.01624	161.26	161.28	96.97	1020.08	1117.05	0.18000	1.7328	1.9128	129
130	2.2237	4.5274	0.01625	157.25	157.27	97.97	1019.50	1117.47	0.18170	1.7289	1.9106	130
131	2.2838	4.6498	0.01625	153.36	153.38	98.97	1018.92	1117.89	0.18339	1.7250	1.9084	131
132	2.3452	4.7750	0.01626	149.58	149.60	99.97	1018.34	1118.31	0.18508	1.7211	1.9062	132
133	2.4081	4.9030	0.01626	145.91	145.93	100.97	1017.76	1118.73	0.18676	1.7172	1.9040	133
134	2.4725	5.0340	0.01626	142.34	142.36	101.97	1017.18	1119.15	0.18845	1.7134	1.9018	134
135	2.5382	5.1679	0.01627	138.87	138.89	102.97	1016.59	1119.56	0.19013	1.7095	1.8996	135
136	2.6055	5.3049	0.01627	135.50	135.52	103.97	1016.01	1119.98	0.19181	1.7056	1.8974	136
137	2.6743	5.4450	0.01628	132.22	132.24	104.97	1015.43	1120.40	0.19348	1.7018	1.8953	137
138	2.7446	5.5881	0.01628	129.04	129.06	105.97	1014.85	1120.82	0.19516	1.6979	1.8931	138
139	2.8165	5.7345	0.01629	125.94	125.96	106.97	1014.26	1121.23	0.19683	1.6942	1.8910	139
140	2.8900	5.8842	0.01629	122.94	122.96	107.96	1013.69	1121.65	0.19850	1.6903	1.8888	140
141	2.9651	6.0371	0.01630	120.01	120.03	108.96	1013.11	1122.07	0.20016	1.6865	1.8867	141
142	3.0419	6.1934	0.01630	117.16	117.18	109.96	1012.52	1122.48	0.20182	1.6828	1.8846	142
143	3.1204	6.3532	0.01631	114.40	114.42	110.96	1011.94	1122.90	0.20348	1.6790	1.8825	143
144	3.2006	6.5164	0.01631	111.70	111.72	111.96	1011.35	1123.31	0.20514	1.6753	1.8804	144
145	3.2825	6.6832	0.01632	109.09	109.11	112.96	1010.77	1123.73	0.20679	1.6715	1.8783	145
146	3.3662	6.8536	0.01632	106.54	106.56	113.96	1010.18	1124.14	0.20845	1.6678	1.8763	146
147	3.4517	7.0277	0.01633	104.06	104.08	114.96	1009.59	1124.55	0.21010	1.6641	1.8742	147
148	3.5390	7.2056	0.01633	101.65	101.67	115.96	1009.01	1124.97	0.21174	1.6604	1.8721	148
149	3.6282	7.3872	0.01634	99.306	99.322	116.96	1008.42	1125.38	0.21339	1.6567	1.8701	149
150	3.7194	7.5727	0.01634	97.022	97.038	117.96	1007.83	1125.79	0.21503	1.6530	1.8680	150
151	3.8124	7.7622	0.01635	94.799	94.815	118.96	1007.24	1126.20	0.21667	1.6493	1.8660	151

[a]Compiled by John A. Goff and S. Gratch.

Table 2. Thermodynamic Properties of Water at Saturation[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
152	3.9074	7.9556	0.01635	92.635	92.651	119.96	1006.66	1126.62	0.21830	1.6457	1.8640	152
153	4.0044	8.1532	0.01636	90.528	90.544	120.97	1006.06	1127.03	0.21994	1.6421	1.8620	153
154	4.1035	8.3548	0.01636	88.477	88.493	121.97	1005.47	1127.44	0.22157	1.6384	1.8600	154
155	4.2046	8.5607	0.01637	86.480	86.496	122.97	1004.88	1127.85	0.22320	1.6348	1.8580	155
156	4.3078	8.7708	0.01637	84.536	84.552	123.97	1004.29	1128.26	0.22482	1.6312	1.8560	156
157	4.4132	8.9853	0.01638	82.642	82.658	124.97	1003.70	1128.67	0.22645	1.6276	1.8540	157
158	4.5207	9.2042	0.01638	80.798	80.814	125.97	1003.11	1129.08	0.22807	1.6239	1.8520	158
159	4.6304	9.4276	0.01639	79.001	79.017	126.97	1002.51	1129.48	0.22969	1.6204	1.8501	159
160	4.7424	9.6556	0.01639	77.251	77.267	127.97	1001.92	1129.89	0.23130	1.6168	1.8481	160
161	4.8566	9.8882	0.01640	75.546	75.562	128.97	1001.33	1130.30	0.23292	1.6133	1.8462	161
162	4.9732	10.126	0.01640	73.885	73.901	129.97	1000.74	1130.71	0.23453	1.6097	1.8442	162
163	5.0921	10.368	0.01641	72.267	72.283	130.98	1000.13	1131.11	0.23614	1.6062	1.8423	163
164	5.2134	10.615	0.01642	70.690	70.706	131.98	999.54	1131.52	0.23774	1.6027	1.8404	164
165	5.3372	10.867	0.01642	69.153	69.169	132.98	998.94	1131.92	0.23935	1.5990	1.8384	165
166	5.4634	11.124	0.01643	67.654	67.670	133.98	998.35	1132.33	0.24095	1.5956	1.8365	166
167	5.5921	11.386	0.01643	66.194	66.210	134.98	997.75	1132.73	0.24255	1.5920	1.8346	167
168	5.7233	11.653	0.01644	64.770	64.786	135.98	997.16	1133.14	0.24414	1.5887	1.8328	168
169	5.8572	11.925	0.01644	63.382	63.398	136.99	996.55	1133.54	0.24574	1.5852	1.8309	169
170	5.9936	12.203	0.01645	62.029	62.045	137.99	995.95	1133.94	0.24733	1.5817	1.8290	170
171	6.1328	12.487	0.01645	60.710	60.726	138.99	995.36	1134.35	0.24892	1.5782	1.8271	171
172	6.2746	12.775	0.01646	59.423	59.439	139.99	994.76	1134.75	0.25051	1.5748	1.8253	172
173	6.4192	13.070	0.01647	58.168	58.184	141.00	994.15	1135.15	0.25209	1.5713	1.8234	173
174	6.5666	13.370	0.01647	56.944	56.960	142.00	993.55	1135.55	0.25367	1.5679	1.8216	174
175	6.7168	13.676	0.01648	55.750	55.766	143.00	992.95	1135.95	0.25525	1.5644	1.8197	175
176	6.8699	13.987	0.01648	54.586	54.602	144.00	992.35	1136.35	0.25683	1.5611	1.8179	176
177	7.0259	14.305	0.01649	53.450	53.466	145.00	991.75	1136.75	0.25841	1.5577	1.8161	177
178	7.1849	14.629	0.01650	52.341	52.357	146.01	991.14	1137.15	0.25998	1.5543	1.8143	178
179	7.3469	14.959	0.01650	51.260	51.276	147.01	990.54	1137.55	0.26155	1.5508	1.8124	179
180	7.5119	15.295	0.01651	50.203	50.220	148.01	989.93	1137.94	0.26312	1.5475	1.8106	180
181	7.6801	15.637	0.01651	49.173	49.190	149.02	989.32	1138.34	0.26468	1.5442	1.8089	181

[a]Compiled by John A Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Concluded)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat Vapor s_g	
182	7.8514	15 986	0 01652	48 168	48 185	150 02	988.72	1138.74	0.26625	1.5408	1 8071	182
183	8.0258	16.341	0.01652	47.187	47 204	151.02	988 12	1139.14	0.26781	1.5375	1.8053	183
184	8.2035	16.703	0.01653	46.229	46 246	152.03	987 50	1139.53	0.26937	1.5341	1.8035	184
185	8.3845	17.071	0.01654	45.294	45 311	153.03	986.89	1139.92	0.27093	1.5308	1.8017	185
186	8.5688	17.446	0 01654	44.381	44.398	154 04	986.28	1140.32	0.27248	1.5275	1.8000	186
187	8.7565	17.829	0 01655	43.489	43.506	155.04	985 67	1140.71	0 27404	1.5242	1.7982	187
188	8 9476	18.218	0 01656	42 619	42 636	156.04	985.07	1141.11	0.27559	1.5209	1 7965	188
189	9 1422	18.614	0 01656	41.769	41.786	157.05	984 45	1141.50	0.27713	1 5176	1.7947	189
190	9.3403	19.017	0 01657	40.939	40 956	158 05	983 84	1141.89	0.27868	1 5143	1.7930	190
191	9.5420	19.428	0 01658	40.128	40.145	159.06	983.22	1142.28	0.28022	1.5111	1.7913	191
192	9.7473	19.846	0 01658	39.337	39.354	160.06	982 61	1142.67	0.28176	1 5078	1 7896	192
193	9 9563	20.271	0 01659	38 563	38 580	161.06	982 00	1143.06	0.28330	1 5045	1 7878	193
194	10 169	20.704	0 01659	37.807	37 824	162.07	981 38	1143 45	0.28484	1 5013	1 7861	194
195	10.386	21.145	0 01660	37.069	37.086	163.08	980 76	1143 84	0.28638	1 4980	1 7844	195
196	10.606	21 594	0 01661	36.348	36 365	164 08	980.15	1144.23	0.28791	1 4949	1.7828	196
197	10.830	22.050	0 01661	35 643	35 660	165 08	979.54	1144 62	0 28944	1.4917	1.7811	197
198	11.058	22 515	0 01662	34.954	34.971	166.09	978.91	1145.00	0.29097	1 4884	1 7794	198
199	11.290	22.987	0 01663	34 281	34 298	167 10	978 29	1145 39	0 29250	1.4852	1.7777	199
200	11.526	23 468	0.01663	33.623	33.640	168 10	977.68	1145.78	0 29402	1.4820	1 7760	200
201	11.767	23.957	0 01664	32.980	32 997	169.11	977.05	1146.16	0.29554	1.4789	1 7744	201
202	12 011	24 455	0 01665	32 351	32 368	170 11	976 43	1146 54	0 29706	1 4756	1 7727	202
203	12.260	24.961	0.01665	31 737	31.754	171.12	975.81	1146.93	0.29858	1.4725	1.7711	203
204	12.513	25 476	0 01666	31 136	31.153	172 12	975.19	1147.31	0 30010	1 4693	1 7694	204
205	12.770	26 000	0 01667	30 549	30 566	173.13	974.56	1147 69	0 30161	1 4662	1 7678	205
206	13.031	26.532	0.01667	29 974	29.991	174.14	973 94	1148.08	0.30312	1.4631	1.7662	206
207	13.297	27 074	0 01668	29 413	29.430	175.14	973.32	1148 46	0.30463	1.4600	1 7646	207
208	13.568	27.625	0.01669	28.863	28.880	176.15	972.69	1148.84	0.30614	1.4568	1.7629	208
209	13.843	28.185	0 01669	28 326	28.343	177.16	972.06	1149.22	0.30765	1.4536	1.7613	209
210	14.123	28 754	0 01670	27 801	27 818	178.17	971.43	1149.60	0.30915	1.4506	1.7597	210
211	14 407	29.333	0 01671	27 287	27 304	179.17	970 81	1149.98	0 31065	1.4474	1.7581	211
212	14.696	29.921	0.01671	26.784	26 801	180 18	970 17	1150.35	0 31215	1.4444	1.7565	212

[a]Compiled by John A. Goff and S. Gratch.

TABLE 3. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE[a]

Abs Press. In. Hg p	Temp F t	Specific Volume		Enthalpy			Entropy			Abs Press. In Hg p
		Sat Liquid v_f	Sat Vapor v_g	Sat Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat Liquid S_f	Evap S_{fg}	Sat Vapor S_g	
0 25	40.23	0.01602	2423.7	8.28	1071 1	1079.4	0 0166	2 1423	2 1589	0 25
0 50	58 80	0 01604	1256 4	26.86	1060 6	1087 5	0 0532	2 0453	2 0985	0 50
0.75	70.43	0.01606	856.1	38.47	1054.0	1092 5	0 0754	1 9881	2 0635	0 75
1.00	79 03	0 01608	652.3	47.05	1049 2	1096 3	0 0914	1 9473	2 0387	1 00
1 5	91.72	0 01611	444.9	59.71	1042 0	1101 7	0.1147	1 8894	2 0041	1.5
2	101 14	0 01614	339 2	69.10	1036.6	1105 7	0.1316	1 8481	1 9797	2
4	125 43	0 01622	176 7	93.34	1022.7	1116 0	0.1738	1 7476	1 9214	4
6	140.78	0 01630	120 72	108.67	1013.6	1122 3	0 1996	1 6881	1.8877	6
8	152.24	0 01635	92 16	120 13	1006.9	1127 0	0 2186	1.6454	1 8640	8
10	161 49	0.01640	74 76	129 38	1001 4	1130 8	0.2335	1.6121	1 8456	10
12	169 28	0 01644	63 03	137 18	996 7	1133 9	0.2460	1 5847	1 8307	12
14	176 05	0.01648	54 55	143.96	992.6	1136.6	0 2568	1 5613	1.8181	14
16	182 05	0 01652	48 14	149.98	988.9	1138.9	0 2662	1 5410	1.8072	16
18	187.45	0 01655	43.11	155.39	985.7	1141.1	0 2746	1.5231	1 7977	18
20	192 37	0 01658	39 07	160 33	982 7	1143 0	0.2822	1 5069	1 7891	20
22	196 90	0 01661	35.73	164 87	979.8	1144 7	0 2891	1.4923	1.7814	22
24	201.09	0 01664	32.94	169 09	977 2	1146 3	0 2955	1 4789	1.7744	24
26	205.00	0 01667	30 56	173 02	974.8	1147 8	0.3014	1 4665	1 7679	26
28	208 67	0.01669	28 52	176 72	972.5	1149 2	0 3069	1.4550	1 7619	28
30	212.13	0.01672	26.74	180.19	970.3	1150 5	0.3122	1.4442	1.7564	30
Lb/Sq In										Lb/Sq In.
14.696	212 00	0.01672	26.80	180 07	970 3	1150 4	0 3120	1.4446	1 7566	14 696
16	216 32	0 01674	24.75	184 42	967 6	1152 0	0 3184	1.4313	1.7497	16
18	222 41	0.01679	22.17	190 56	963 6	1154 2	0.3275	1 4128	1.7403	18
20	227.96	0.01683	20.089	196.16	960 1	1156.3	0.3356	1.3962	1.7319	20
22	233 07	0.01687	18.375	201 33	956 8	1158.1	0 3431	1.3811	1 7242	22
24	237 82	0.01691	16.938	206 14	953.7	1159 8	0 3500	1 3672	1.7172	24
26	242.25	0 01694	15.715	210 62	950 7	1161 3	0 3564	1.3544	1 7108	26
28	246.41	0 01698	14.663	214 83	947.9	1162.7	0.3623	1 3425	1.7048	28
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	30
32	254 05	0.01704	12.940	222.59	942.8	1165 4	0 3733	1.3209	1 6941	32
34	257 58	0.01707	12.226	226.18	940.3	1166 5	0.3783	1 3110	1.6893	34
36	260 95	0.01709	11.588	229 60	938.0	1167 6	0 3831	1 3017	1.6848	36
38	264 16	0 01712	11.015	232.89	935.8	1168 7	0 3876	1.2929	1.6805	38
40	267.25	0 01715	10 498	236 03	933.7	1169.7	0.3919	1 2844	1 6763	40
42	270.21	0 01717	10.029	239.04	931.6	1170 7	0.3960	1 2764	1.6724	42
44	273.05	0 01720	9 601	241.95	929.6	1171.6	0 4000	1 2687	1 6687	44
46	275 80	0 01722	9 209	244.75	927.7	1172 4	0.4038	1.2613	1.6652	46
48	278 45	0 01725	8.848	247 47	925.8	1173.3	0.4075	1 2542	1 6617	48
50	281.01	0 01727	8 515	250.09	924 0	1174.1	0 4110	1.2474	1.6585	50
52	283.49	0 01729	8.208	252.63	922.2	1174.8	0.4144	1.2409	1.6553	52
54	285 90	0 01731	7 922	255 09	920.5	1175 6	0 4177	1.2346	1 6523	54
56	288 23	0 01733	7 656	257 50	918.8	1176 3	0.4209	1.2285	1.6494	56
58	290.50	0 01736	7.407	259 82	917.1	1176 9	0.4240	1.2226	1.6466	58
60	292.71	0 01738	7 175	262.09	915.5	1177 6	0.4270	1.2168	1.6438	60
62	294.85	0 01740	6 957	264 30	913 9	1178.2	0.4300	1.2112	1.6412	62
64	296 94	0 01742	6.752	266.45	912.3	1178.8	0.4328	1.2059	1.6387	64
66	298 99	0 01744	6 560	268 55	910 8	1179.4	0.4356	1 2006	1.6362	66
68	300 98	0 01746	6.378	270 60	909.4	1180.0	0.4383	1.1955	1.6338	68
70	302.92	0 01748	6.206	272.61	907 9	1180.6	0 4409	1.1906	1.6315	70
72	304.83	0.01750	6 044	274.57	906 5	1181.1	0.4435	1 1857	1.6292	72
74	306 68	0 01752	5.890	276.49	905.1	1181.6	0.4460	1.1810	1.6270	74
76	308.50	0 01754	5.743	278.37	903.7	1182.1	0.4484	1 1764	1.6248	76
78	310 29	0 01755	5.604	280 21	902 4	1182 6	0 4508	1.1720	1 6228	78
80	312 03	0.01757	5.472	282 02	901 1	1183 1	0 4531	1.1676	1 6207	80
82	313 74	0.01759	5.346	283 79	899.7	1183.5	0 4554	1 1633	1.6187	82
84	315.42	0.01761	5 226	285 53	898 5	1184 0	0 4576	1 1592	1 6168	84
86	317 07	0.01762	5.111	287.24	897.2	1184.4	0 4598	1.1551	1.6149	86
88	318 68	0.01764	5.001	288.91	895 9	1184 8	0 4620	1.1510	1 6130	88
90	320.27	0 01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1 6112	90
92	321.83	0 01768	4.796	292.18	893 5	1185 7	0.4661	1.1433	1 6094	92
94	323 36	0 01769	4.699	293.78	892.3	1186 1	0 4682	1.1394	1 6076	94
96	324 87	0 01771	4.606	295.34	891.1	1186.4	0 4702	1.1358	1 6060	96
98	326 35	0 01772	4.517	296 89	889.9	1186 8	0.4721	1 1322	1 6043	98
100	327 81	0 01774	4.432	298 40	888.8	1187.2	0.4740	1.1286	1 6026	100
150	358 42	0 01809	3.015	330 51	863.6	1194.1	0.5138	1 0556	1.5694	150
200	381.79	0 01839	2.288	355.36	843.0	1198.4	0.5435	1 0018	1 5453	200
300	417 33	0 01890	1 5433	393 84	809.0	1202.8	0 5879	0.9225	1.5104	300
400	444 59	0 0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	400
500	467 01	0 0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	500

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necessary to add a correction term $\bar{h}$ as follows,

$$\bar{h} = \frac{\mu(1 - \mu)B}{1 + aW_s\,\mu} \tag{5a}$$

In Table 4 are listed values of the coefficient B and maximum values of the correction term $\bar{h}$, the latter occurring at the same values of μ as do those of the correction term $\bar{v}$.

Unfortunately, values of the entropy s of moist air per pound of dry air computed from the simple relation,

$$s = s_a + \mu s_{as} \tag{6}$$

do not approximate the true values as closely as might be desired except at temperatures considerably lower than 150 F. Only a relatively small portion of the error is contributed by the correction term

$$\bar{s} = \frac{\mu(1 - \mu)C}{1 + aW_s\,\mu} \tag{6a}$$

Table 4 lists values of the coefficient C and maximum values of the correction term $\bar{s}$, the latter occurring at the same values of μ as do those of the correction terms $\bar{v}$ and $\bar{h}$. The larger portion of the error is contributed by the so-called *mixing entropy*. It can be expressed as an additional correction term $\bar{\bar{s}}$ as follows,

$$\begin{aligned}\bar{\bar{s}} = 0.1579\,[&(1 + \mu aW_s)\log_{10}(1 + \mu aW_s) - \mu aW_s\log_{10}(\mu) \\ &- \mu(1 + aW_s)\log_{10}(1 + aW_s)]\end{aligned} \tag{6b}$$

Maximum values of $\bar{\bar{s}}$ and the values of μ at which they occur are given in Table 4. In Equation 6b $\log_{10}$ denotes logarithm to the base 10.

MOLLIER DIAGRAM

It is a fundamental proposition of thermodynamics that *when a fluid flows across a section fixed in space it convects with it an amount of energy equal to its enthalpy as determined by the pressure, temperature, and composition of the fluid at that section.* This fundamental proposition provides the correct procedure for applying the law of conservation of energy to

TABLE 4. COEFFICIENTS A, B, C APPEARING IN EQUATIONS 4a, 5a, 6a, MAXIMUM VALUES OF CORRECTIONS DEFINED BY EQUATIONS 4a, 5a, 6a. DEGREE OF SATURATION AT WHICH THESE MAXIMA OCCUR, $\bar{\mu}_m$. MAXIMUM VALUE OF CORRECTION DEFINED BY EQUATION 6b. DEGREE OF SATURATION AT WHICH THIS MAXIMUM OCCURS, $\bar{\bar{\mu}}_m$.

(Standard Atmospheric Pressure)

t (F)	A (ft^3/lb_a)	B (Btu/lb_a)	C ($Btu/F/lb_a$)	$\bar{v}_{max}$ (ft^3/lb_a)	$\bar{h}_{max}$ (Btu/lb_a)	$\bar{s}_{max}$ ($Btu/F/lb_a$)	$\bar{\mu}_m$	$\bar{\bar{s}}_{max}$ ($Btu/F/lb_a$)	$\bar{\bar{\mu}}_m$
96	0.0018	0.0286	0.00004	0.0004	0.0069	0.00001	0.4925	0.0015	0.3650
112	0.0042	0.0650	0.00009	0.0010	0 0155	0.00002	0.4878	0.0025	0.3632
128	0.0096	0.1439	0.00020	0.0022	0.0332	0.00005	0.4805	0.0040	0 3602
144	0.0215	0.3149	0.00042	0.0047	0.0693	0.00009	0.4691	0.0065	0.3557
160	0.0487	0.6969	0.00091	0 0099	0.1418	0.00019	0.4511	0.0106	0.3485
176	0.1169	1.636	0.00207	0.0207	0.2903	0.00037	0.4213	0.0179	0.3363
192	0.3363	4.608	0.00567	0.0451	0.6180	0.00076	0.3662	0.0333	0.3129

the processes occurring most frequently in air conditioning practice. Thus if moist air is flowing through a duct it carries across any section of the duct energy of amount mh Btu per minute and water of amount mW pounds per minute, if m denotes the weight of dry air crossing the section per minute.

The foregoing considerations suggest the importance of having accurate knowledge regarding the enthalpy of the fluid in question and the desirability of using enthalpy as one of the coordinates in graphical representation. The use of enthalpy h and humidity ratio W as coordinates in the case of moist air is due to Mollier [4]. A convenient modification of the Mollier diagram introduced by Goff is obtained by taking humidity ratio W as ordinate and reduced enthalpy (h-1000W) as abscissa. A Mollier Diagram modified in this way is enclosed in the envelope attached to the inside back cover and an abridgement of the Diagram is shown in Fig. 1.

The Mollier Diagram is a constant-pressure chart, the one provided with this book being drawn for standard atmospheric pressure from the data in Table 1. Along the axis of abscissae ($W = 0$, $\mu = 0$) are plotted values of the specific enthalpy of dry air h_a at one-degree intervals of temperature. Values of humidity ratio at saturation W_s plotted against values of reduced enthalpy at saturation (h_s-1000W_s) determine the saturation curve ($\mu = 100$ per cent). Lines of constant temperature connect points on the saturation curve with corresponding points on the dry-air axis and are inclined upward to the right. They are drawn straight in accordance with Equations 3 and 5 because the curvature contributed by the correction term 5a is inappreciable at all temperatures within the range of the chart. The portion of each isotherm lying between the dry-air axis and the saturation curve is divided into 10 equal parts by curves of constant per cent saturation. The per cent saturation of any point below the saturation curve is readily determined by linear interpolation along the isotherm through that point.

Each isotherm breaks at the saturation curve to incline upward to the left into the two-phase region above the saturation curve. The ordinate of a point in this region is the total weight of water in both the vapor phase (moist air) and the condensed phase (liquid or solid) per pound of dry air in both phases. Neglecting the very small amount of dissolved air in the condensed phase, it is the weight of water in both phases per pound of dry air in the vapor phase. The ordinate at the break in the isotherm through the point in question is the weight of water per pound of dry air in the vapor phase. Consequently, the difference between the two ordinates is the weight of condensed phase per pound of dry air in the vapor phase.

It has been stated that the region above the saturation curve is the two-phase region. This is so except in the wedge with apex on the saturation curve at 32 F where three distinct phases, namely, solid, liquid, and vapor coexist. In fact, this wedge separates the liquid-vapor region above the wedge from the solid-vapor region below it. A point inside the wedge divides the horizontal line extending through it from one boundary to the other into two segments which are in the same ratio as are the weights of solid and liquid. The temperature is 32 F throughout the wedge.

The isotherms in the two-phase regions above the saturation curve have been extended downward to the right into the vapor-phase region

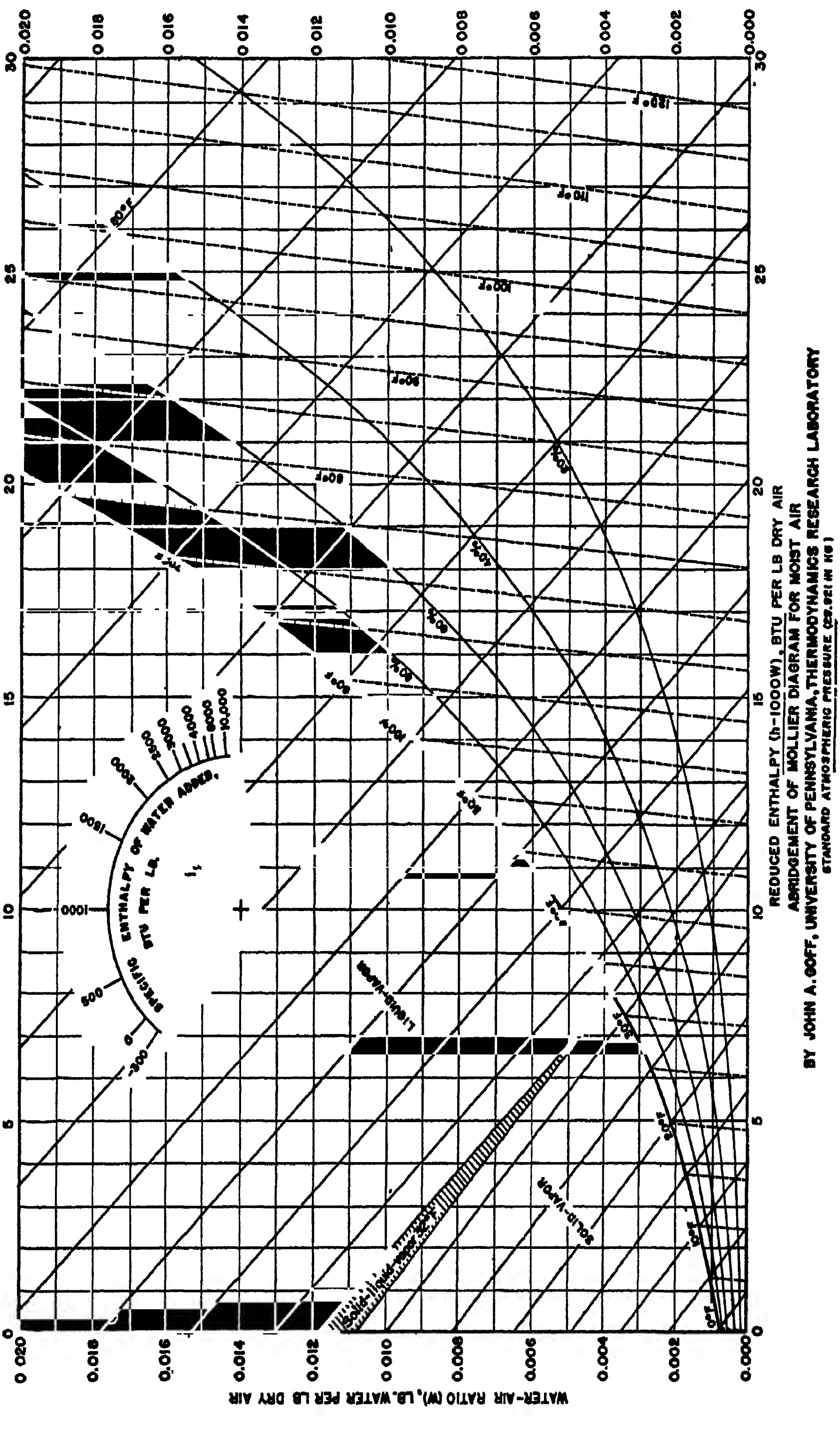

FIG. 1. ABRIDGEMENT OF MOLLIER DIAGRAM FOR MOIST AIR

below the saturation curve as lines of constant *thermodynamic wet-bulb temperature.* The definition of thermodynamic wet-bulb temperature will be given later.

On the Mollier Diagram provided with THE 1947 GUIDE there has been drawn a *protractor* from which can be determined the direction in which the state point of a mixture of water and dry air will be moved *by simultaneous addition of energy and water without addition of dry air.* A particular direction is specified by the numerical value of the *ratio of energy to water added* which ratio is designated as q and called the *specific enthalpy of water added, Btu per pound.* The protractor is useful in locating the *condition line* of a cooling load or heating load problem.

DERIVED PROPERTIES

Thermodynamic Wet-bulb Temperature. For any state of moist air there exists a temperature t^* at which liquid (or solid) water may be evaporated into the air to bring it to saturation at exactly this same temperature. The humidity ratio of the air is increased from a given initial value W to the value W_s^* corresponding to saturation at the temperature t^*; the enthalpy of the air is increased from a given initial value h to the value h_s^* corresponding to saturation at the temperature t^*; the weight of water added per pound of dry air is $W_s^* - W$ and this adds energy of amount $(W_s^* - W)\, h_w^*$, where h_w^* denotes the specific enthalpy of the water as added at the temperature t^*; therefore, if the process is strictly adiabatic,

$$h + (W_s^* - W)h_w^* = h_s^* \tag{7}$$

The solution of Equation 7 for given values of h and W is called *thermodynamic wet-bulb temperature.*

Example 1. Find the thermodynamic wet-bulb temperature of moist air at 80 F, 50 per cent saturation, atmospheric pressure.

Solution. From the data of Table 1, the enthalpy of the air is $h = 19.221 + 0.50 \times 24.47 = 31.46$ Btu/lb$_a$ (Equation 5). To a first approximation this is the enthalpy at saturation at the thermodynamic wet-bulb temperature which is therefore approximately 67 F.

At 67 F the humidity ratio at saturation is 0.01424 lb$_w$/lb$_a$ and the specific enthalpy of liquid water is 35.11 Btu/lb$_w$. The humidity ratio of the air is $W = 0.50 \times 0.02233 = 0.01117$ lb$_w$/lb$_a$ (Equation 3). Therefore, to a second approximation, the enthalpy at saturation at the thermodynamic wet-bulb temperature is $h_s^* = 31.46 + (0.01424 - 0.01117) \times 35.11 = 31.57$ Btu/lb$_a$, Equation 7. Interpolation in Table 1 gives as final answer,

$$t^* = 66.94 \text{ F}$$

The answer can also be read directly on the Mollier Diagram at the intersection of the 80 F dry-bulb and 50 per cent saturation lines.

The psychrometer is an instrument consisting of two thermometers one of which has the bulb covered with a suitable wick that has been dipped in liquid water and thoroughly wetted by it. On placing the wet-bulb of the instrument in an air stream, the liquid begins to evaporate from the wick and it is usually assumed that such evaporation brings the air immediately adjacent to the wick to saturation. At first this air may reach saturation at a higher or lower temperature than that of the liquid on the wick; but in a relatively short time the temperature of the liquid will have changed to approach equality with that of the air touching the wick, even if this requires the liquid to freeze on the wick. Then the liquid (or solid) will continue for a time to evaporate into the air stream

at this same temperature. This equilibrium temperature is called *wet-bulb temperature.*

It is clear that the readings of an actual wet-bulb thermometer cannot be regarded as values of a thermodynamic property of moist air; for these readings are importantly affected by a number of non-thermodynamic factors including design, construction, installation, and technique of using the instrument. Thus, unless the wet-bulb is effectively shielded against radiation from relatively warm surfaces the process will not be strictly adiabatic as tacitly assumed in writing Equation 7. Also, partial drying of the wick will prevent the air immediately adjacent to it from reaching complete saturation as assumed in Equation 7. A working theory developed by Arnold [5] enables the calculation of corrections to be applied to the readings of the actual instrument in order to make them agree with the values of temperature calculated from Equation 7. Fortunately, and indeed fortuitously, these corrections can be made small, but to emphasize the necessity of making them in accurate experimentation, the temperature defined by Equation 7 is called *thermodynamic wet-bulb temperature.*

Example 2. Find the degree of saturation of moist air at 90 F dry-bulb, 63 F thermodynamic wet-bulb, atmospheric pressure.

Solution. Inserting numerical data from Table 1 into Equation 7 gives

$$(21.625 + 34.31\mu) + (0.01235 - 0.03118\mu) \times 31.12 = 28.57$$

The solution of this equation is direct and the final answer is

$$\mu = 19.67 \text{ per cent}$$

The per cent saturation may also be read directly at intersection of 90 F dry-bulb and 63 F thermodynamic wet-bulb lines on the Mollier Diagram.

Example 3. Find the temperature to which moist air initially saturated at 40 F and at standard atmospheric pressure must be heated in order to have a thermodynamic wet-bulb temperature of 60 F.

Solution. On the Mollier Diagram follow a horizontal line from the saturation curve at 40 F to its intersection with the 60 F thermodynamic wet-bulb line and read the corresponding temperature directly.

Inserting numerical data from Table 1 into Equation 7, this becomes

$$h_a + 0.005213h_{as}/W_s = 26.46 - (0.01108 - 0.005213) \times 28.12 = 26.295$$

At 85 F the lefthand member of this equation has the value 26.147; at 86 F its value is 26 389; by linear interpolation the answer is: $t = 85.61$ F.

Dew-Point Temperature. Corresponding to any given state of moist air there exists another state on the saturation curve having the same humidity ratio W and same pressure p as the given state. The temperature at this other state on the saturation curve is called the *dew-point temperature* of the given state. Obviously, if moist air is cooled at constant pressure and constant humidity ratio it will reach saturation when its temperature falls to a value equal to its dew-point temperature. This will usually be marked by the first appearance of a coexisting condensed phase. In one type of dew-point apparatus a continuous sample of air is passed over a mirror which can be cooled by external refrigeration and whose temperature can be accurately measured. The measured temperature at which the intensity of light reflected from the mirror is abruptly diminished by condensation is taken to be the dew-point temperature of the air sample. Examples 4 and 5 illustrate the relation between the dew-point, degree of saturation and dry-bulb temperature.

Example 4. Find the dew-point temperature of moist air at 80 F, 50 per cent saturation, atmospheric pressure.

Solution. On the Mollier Diagram follow a horizontal line from a given state point (80 F, 50 per cent) to the saturation curve and read the temperature at the intersection.

To solve from Table 1:

From the data in Table 1, the humidity ratio of the air is $W = 0.50 \times 0.02233 = 0.01117$ lb_w/lb_a. By interpolation this is found to be the humidity ratio at saturation at 60.22 F which is therefore the required answer.

Example 5. Find the degree of saturation of moist air at 90 F dry-bulb, 40 F (dew-point), atmospheric pressure.

Solution. On the Mollier Diagram follow a horizontal line from 40 F on the saturation curve to the 90 F isotherm (dry-bulb) and read the degree of saturation directly.

To solve from Table 1:

From the data in Table 1, the humidity ratio of the air must be $W = 0.005213$. But the humidity ratio at saturation at 90 F is 0.03118; hence the degree of saturation is

$$\mu = 0.005213/0.03118 = 16.72 \text{ per cent}$$

TYPICAL AIR CONDITIONING PROCESSES

The use of Table 1 and the Mollier Diagram in analyzing typical air conditioning processes is best explained by means of illustrative examples. In each of the following, it is to be understood that the process in question takes place at a constant pressure of 29.921 in. Hg, or standard atmospheric pressure.

Heating

The process of adding heat to moist air is represented by a horizontal line on the Mollier Diagram. The length of the line between the initial and final state points is the increase of reduced enthalpy; but, since the humidity ratio is constant, it is also the increase of enthalpy itself and therefore the quantity of heat added per pound of dry air.

Example 6. Air initially at 20 F, 80 per cent saturation is heated to 120 F. Find the quantity of heat required to process 20,000 cfm of heated air.

Solution. From the data in Table 1: the initial humidity ratio is $0.80 \times 0.002152 = 0.001722$ lb_w/lb_a; the initial enthalpy is $4.804 + 0.80 \times 2.302 = 6.646$ Btu/lb_a; the final degree of saturation is $0.001722/0.08149 = 2.113$ per cent; the final enthalpy is $28.841 + 0.02113 \times 90.70 = 30.757$ Btu/lb_a.

It may be supposed that the air is heated between two sections of a duct. The quantities of energy convected across the two sections per pound of dry air crossing them are the two enthalpies calculated. Conservation of energy requires that the difference between these two enthalpies be the quantity of heat added; thus,

$$_Aq_B = 30.757 - 6.646 = 24.111 \text{ Btu/lb}_a.$$

The final volume is $14.611 + 0.02113 \times 1.905 = 14.651$ cu ft/lb_a. Since 20,000 cfm of heated air is to be processed, the total quantity of heat required is

$$_AQ_B = 24.111 \times 20{,}000/14.651 = 32{,}914 \text{ Btu per minute.}$$

On the Mollier Diagram the process is represented by the horizontal line AB, Fig. 2, whose length is the quantity of heat added per pound of dry air. The reduced enthalpy at A is 4.92 while that at B is 29.03, both being read directly from the chart. Since humidity ratio is constant the difference between these reduced enthalpies is also the difference between the enthalpies themselves, namely, 24.11 Btu/lb_a.

Cooling

The process of cooling moist air is also represented by a horizontal line on the Mollier Diagram. The line may extend across the saturation curve into the two-phase region, nevertheless, the length of the line between

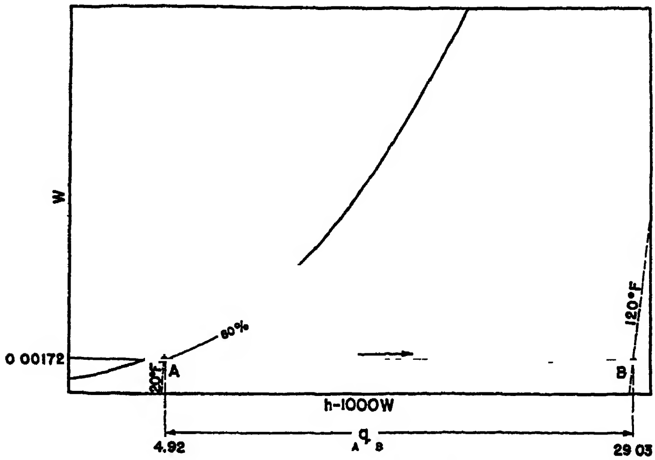

FIG. 2. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 6

the initial and final states is the quantity of heat removed, or refrigeration supplied, per pound of dry air. By following the final isotherm downward to the right to the saturation curve and reading the ordinate there, the weight of water vapor per pound of dry air in the vapor phase is determined. The difference between the initial humidity ratio and this ordinate is the weight of condensed phase per pound of dry air in the final state.

Example 7. Air at 95 F and 50 per cent saturation is cooled to 70 F. Find the refrigeration required to process 20,000 cfm of uncooled air.

Solution. From the data in Table 1: the initial humidity ratio is $0.50 \times 0.03673 = 0.01837$ lb_w/lb_a; the initial enthalpy is $22.827 + 0.50 \times 40.49 = 43.072$ Btu/lb_a; the humidity ratio at saturation at the final temperature is 0 01582 lb_w/lb_a; the quantity of liquid formed is $0.01837 - 0.01582 = 0.00255$ lb_w/lb_a; the enthalpy of the final two-phase mixture is $34.09 + 0.00255 \times 38.11 = 34.187$ Btu/lb_a.

It may be supposed that the air is cooled between two sections of a duct. The quantities of energy convected across the two sections per pound of dry air crossing them

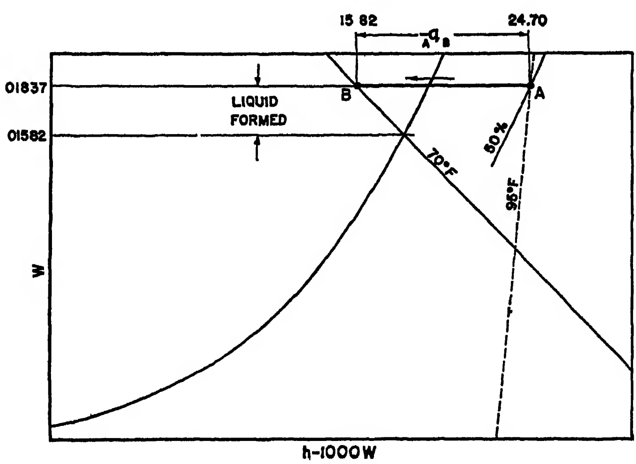

FIG. 3. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 7

are the two enthalpies calculated. Conservation of energy requires that the difference between these two enthalpies be the quantity of heat removed, or refrigeration supplied, between the two sections. Therefore,

$$-_Aq_B = 43{,}072 - 34.187 = 8.885 \text{ Btu/lb}_a$$

The initial volume is 13.980 + 0 50 × 0 822 = 14.391 cu ft/lb$_a$. Since 20,000 cfm of air is to be processed, the total refrigeration required is

$$-_AQ_B = 8.885 \times 20{,}000 \div 14.391 = 12{,}348 \text{ Btu per minute}$$

On the Mollier Diagram the process is represented by the horizontal line AB, Fig. 3, whose length is the quantity of refrigeration required per pound of dry air.

Adiabatic Mixing of Two Air Streams

A typical air conditioning process requiring special analysis is the adiabatic mixing of two air streams. Referring to Fig. 4, let m_1, m_2, m_3 denote the weights of dry air convected across sections F$_1$, F$_2$, F$_3$, respectively, per minute. Then m_1W_1, m_2W_2, m_3W_3 and m_1h_1, m_2h_2, m_3h_3 will

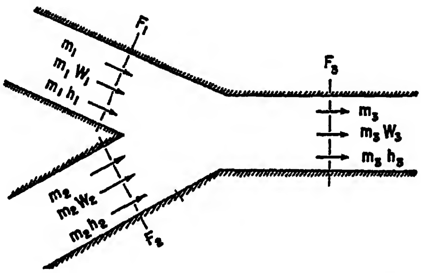

Fig. 4. Adiabatic Mixing of 2 Air Streams

denote the weights of water and the quantities of energy similarly convected. If the mixing is adiabatic, it must be governed by the three equations,

$$\begin{aligned} m_1 + m_2 &= m_3 \\ m_1W_1 + m_2W_2 &= m_3W_3 \\ m_1h_1 + m_2h_2 &= m_3h_3 \end{aligned} \tag{8}$$

Elimination of m_3 gives,

$$\frac{h_2 - h_3}{h_3 - h_1} = \frac{W_2 - W_3}{W_3 - W_1} = \frac{m_1}{m_2} \tag{9}$$

according to which: *on the Mollier Diagram the state point of the resulting mixture lies on the straight line connecting the state points of the two streams being mixed and divides the line into two segments which are in the same ratio as are the weights of dry air in the two streams.*

Example 8. Outside Air at 0 F and 80 per cent saturation is to be mixed adiabatically with recirculated Inside Air at 70 F and 20 per cent saturation in the ratio of one pound of dry air in the former to seven in the latter. Find the temperature and degree of saturation of the resulting mixture.

Solution. The humidity ratio W_3 and the enthalpy h_3 of the resulting mixture must satisfy Equations 9, namely,

$$\frac{0.003164 - W_3}{W_3 - 0.000630} = \frac{20.270 - h_3}{h_3 - 0.668} = \frac{1}{7}$$

from which: $W_3 = 0.002847$, $h_3 = 17.820$. At the temperature of the resulting mixture, therefore,

$$h_a + 0.002847\ h_{as}/W_s = 17.820$$

At 61 F the lefthand member of this equation has the value 17 750; at 62 F its value is 17.991; by interpolation the temperature of the resulting mixture is 61.29 F where the humidity ratio at saturation, also by interpolation, is 0 01161; hence the degree of saturation of the resulting mixture is

$$\mu = 0.002847 \div 0.01161 = 24.52 \text{ per cent}$$

On the Mollier Diagram, Fig. 5, a straight line is drawn between point 1 (0 F, 80 per cent) and point 2 (70 F, 20 per cent); then point 3 is located on the line one-eighth of the distance from point 2 to point 1. The temperature and degree of saturation at point 3 are read directly.

Adiabatic Mixing with Injected Water

Another typical air conditioning process is that of injecting water into an air stream to mix with it adiabatically. Let $W_2 - W_1$, denote the increase in humidity ratio of the air; this is obviously the quantity of water injected per pound of dry air; it follows that the quantity of energy injected per pound of dry air is $(W_2 - W_1)h_w$, where h_w denotes the specific enthalpy of the water as injected; if the process is adiabatic this produces an equal increase in the enthalpy of the air, namely, $h_2 - h_1$; therefore,

$$h_2 - h_1 = h_w(W_2 - W_1) \tag{10}$$

according to which: *the process of injecting water into an air stream to mix adiabatically with it is represented by a straight line on the Mollier Diagram whose direction is fixed by the specific enthalpy of the water as injected.* The protractor drawn on the Mollier Diagram provided with this book provides a convenient means for determining this direction.

Example 9. It is desired to increase the humidity ratio of air at 70 F dry-bulb, without changing its temperature. Under what conditions may water be injected in order to accomplish the desired result?

Solution. At 70 F the increase of enthalpy per unit increase of humidity ratio is $h_{as}/W_s = 17.27 \div 0.01582 = 1092$ Btu per pound of water. This must be the specific enthalpy of the water added if the state point of the air is to be moved along the 70 F isotherm. Saturated steam at 668 F has this specific enthalpy[a].

On the Mollier Diagram, Fig. 6, it is seen that the 70 F isotherm is parallel to the line on the protractor for a specific enthalpy of 1092 Btu per pound.

Adiabatic Saturation

Any process by which the state point of moist air is moved to the saturation curve adiabatically may properly be called adiabatic saturation.

Example 10. Liquid water chilled to 35 F is evaporated into an air stream initially at 90 F and 50 per cent saturation. How much water must be evaporated to bring the air to saturation at what temperature?

Solution. The initial enthalpy of the air is $21.625 + 0.50 \times 34.31 = 38.780$ Btu/lb$_a$; the initial humidity ratio is $0.50 \times 0.03118 = 0.01559$ lb$_w$/lb$_a$; the specific enthalpy of the chilled water is 3.06 Btu/lb$_w$; therefore, the temperature at which the air reaches the saturation curve must be such that the enthalpy h_s and humidity ratio W_s at saturation satisfy the equation,

$$h_s - (W_s - 0.01559) \times 3.06 = 38.780$$

The solution is 75.19 F where the humidity ratio at saturation is 0.01894; consequently,

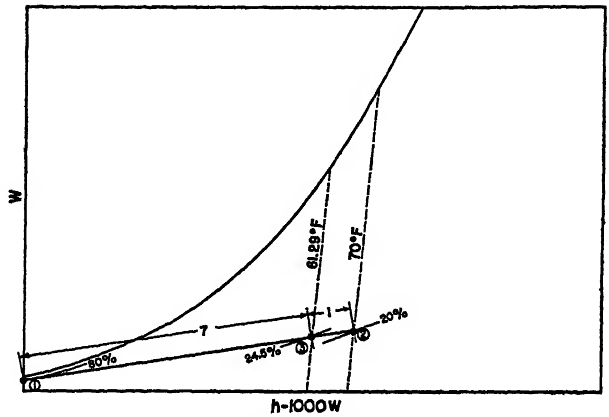

Fig. 5. Illustration of Use of Mollier Diagram in Solution of Example 8

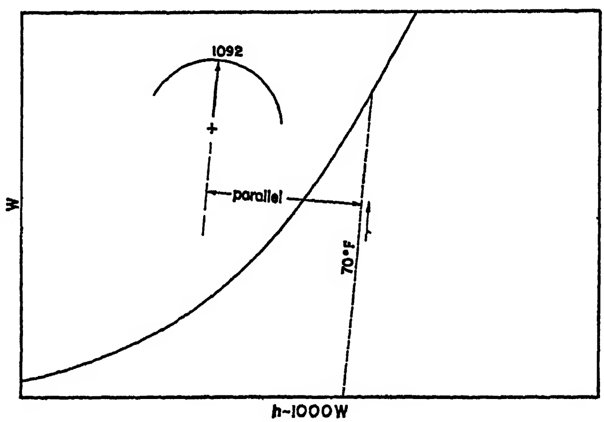

Fig. 6. Illustration of Use of Mollier Diagram in Solution of Example 9

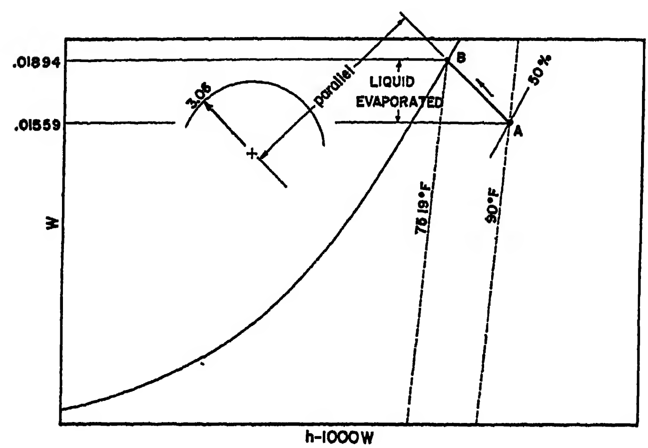

Fig. 7. Illustration of Use of Mollier Diagram in Solution of Example 10

the weight of water evaporated is 0.01894 − 0.01559 = 0.00335 lb per pound of dry air.

On the Mollier Diagram, Fig. 7, a line is drawn through the initial state point (90 F, 50 per cent saturation) in the direction given by the protractor for a specific enthalpy of 3.06 Btu/lb_W.

If air is saturated adiabatically with spray water which is recirculated, the water will ultimately assume a temperature such that the air is brought to saturation at exactly the same temperature; that is, the water will assume the thermodynamic wet-bulb temperature of the air.

Example 11. Air at 75 F and 60 per cent saturation is saturated adiabatically with recirculated spray water. Find the resulting temperature and the weight of water added per pound of dry air.

Solution. In view of the foregoing remarks the solution of this example reduces to the determination of the thermodynamic wet-bulb temperature of the air. Its humidity

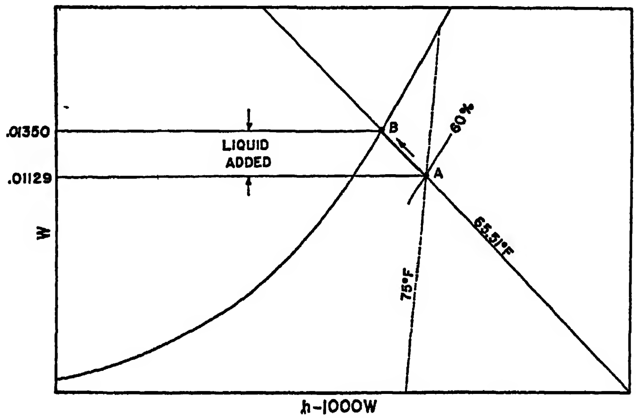

FIG. 8. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 11

ratio is 0.60 × 0.01882 = 0.01129; its enthalpy is 18 018 + 0 60 × 20 59 = 30.372; Equation 7 defining thermodynamic wet-bulb temperature becomes

$$h_s^* - (W_s^* - 0.01129)h_W^* = 30.372$$

At 65 F the value of the lefthand member is 29.995; at 66 F its value is 30.746; by interpolation the thermodynamic wet-bulb temperature is 65.51 F where the humidity ratio at saturation is 0 01350; consequently the weight of water added is 0.01350 − 0.01129 = 0.00221 lb per pound of dry air.

On the Mollier Diagram, Fig. 8, the process is represented by the line AB which is a segment of the 65.51 F thermodynamic wet-bulb line. The difference between the ordinates at B and at A is the weight of water added per pound of dry air.

Cooling Load

The problem of calculating the cooling load for an air conditioned space usually reduces to the determination of the quantity of *inside air* that must be withdrawn and the condition to which it must be brought by suitable processing so that its return to the conditioned space will have the net effect of removing given amounts of energy and water from the space.

Let M denote the weight of dry air withdrawn with *inside air* per hour. With it will be withdrawn energy of amount Mh_1 and water of amount

MW_1 per hour, where h_1 and W_1 denote the enthalpy and humidity ratio of the *inside air*, respectively. The weight of dry air returned with the *conditioned air* will necessarily be the same as that withdrawn with the *inside air*, but with it must be returned a smaller quantity of energy Mh and a smaller quantity of water MW. Let ΔQ and ΔW denote the given amounts of energy and water to be removed from the conditioned space per hour; then

$$Mh = Mh_1 - \Delta Q$$
$$MW = MW_1 - \Delta W$$

Eliminating M and letting q denote the ratio of energy removed to water removed, that is, $q = \Delta Q/\Delta W$,

$$\frac{h - h_1}{W - W_1} = q \qquad (11)$$

according to which: *all possible states for the conditioned air lie on a straight line on the Mollier Diagram passing through the state point of the inside air in the direction specified by the numerical value of the ratio q.* This line is called the *condition line* for the given problem. If the condition line crosses the saturation curve, the point of intersection is called the *apparatus dew-point* for the given problem.

The protractor on the Mollier Diagram facilitates the drawing of the condition line and the locating of the apparatus dew-point. For this purpose the numerical value of the ratio q is to be regarded as a value of the *specific enthalpy of water added, Btu per pound.*

Example 12. A condition of 80 F dry-bulb, and 67 F thermodynamic wet-bulb, is to be maintained in a clothing store, outside conditions being 95 F dry-bulb, and 75 F thermodynamic wet-bulb. The energy gain from normal heat transmission is estimated at 16,000 Btu per hour, that from solar radiation at 48,000 Btu per hour. The energy generated by lights, fans, etc. is estimated at 13,900 Btu per hour. The ventilation requirement is 30,000 cu ft per hour. The number of occupants is 50. Find the apparatus dew-point.

Solution. The properties of *inside air* and *outside air* are readily calculated from the data in Table 1, see especially Example 2.

		Inside Air	Outside Air
μ	=	0.5024	0.3848
h	=	31.514	38.408
W	=	0.01122	0.01413
v	=	....	14.296

The weight of dry air entering with the *ventilating air* is 30,000/14.296 = 2098.5 lb per hour which brings with it energy of amount 2098.5 × 38.408 = 80.595 Btu per hour and water of amount 2098.5 × 0.01413 = 29.659 lb per hour.

The weight of dry air displaced from the store by the *ventilating air* is 2098.5 lb per hour which takes with it energy of amount 2098.5 × 31.514 = 66,132 Btu per hour and water of amount 2098.5 × 0.01122 = 23.541 lb per hour.

Each occupant may be regarded as a normal person standing at rest and evaporating (1386 grains) 0.198 lb of water per hour (value obtained by interpolation between Curves D and C Fig. 7, Chapter 12) at about 79 F. From this source there is water of amount 50 × 0.198 = 9 90 lb per hour and energy of amount 9.90 × 1095.7 = 10,847 Btu per hour added to the conditioned space. In addition each occupant loses 225 Btu per hour by conduction, convection, and radiation, making a total for 50 persons of 11,300 Btu per hour.

The net energy gain is 16,000 + 48,000 + 13,900 + 80,595 − 66,132 + 10,847 + 11,300 = 114,510 Btu per hour. The net water gain is 29.659 − 23.541 + 9.90 = 16.018 lb per hour. Accordingly the direction of the condition line is fixed by the ratio, q = 114,510 ÷ 16.018 = 7148.8 Btu per pound of water.

On the Mollier Diagram, Fig. 9, the direction of the condition line is given by the protractor for a *specific enthalpy of water added* of 7148.8 Btu per pound. The line itself

passes through the state point of the inside air and intersects the saturation curve at the apparatus dew-point.

According to Equation 11 the enthalpy h_s and humidity ratio W_s at the apparatus dew-point must satisfy the equation

$$7148.8W_s - h_s = 7148.8 \times 0.01122 - 31.514 = 48.681$$

At 58 F the left-hand member has the value 48.513; at 59 F its value is 50.641; by interpolation the apparatus dew-point is 58.08 F.

It would be a mistake to assume that the refrigeration to be supplied is equal to the net energy to be removed; for in general water is to be removed simultaneously and unless this is removed as liquid at 32 F it will automatically take some energy with it. Thus, unless the water is removed as solid (ice) the refrigeration to be supplied will be somewhat less than the net energy to be removed.

Example 13. Referring to the cooling load problem of Example 12, suppose that the conditioning process consists of cooling a portion of the *inside air* to the apparatus dew-point temperature, separating out the liquid thus formed, and returning the resulting saturated mixture to the conditioned space. Find the quantity of *inside air* that must be processed in this manner and the corresponding quantity of refrigeration required.

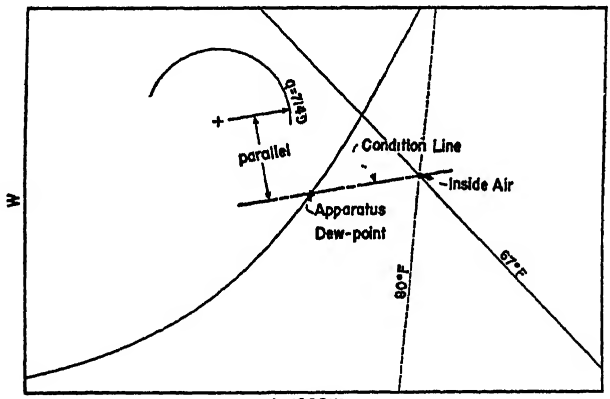

FIG. 9. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 12

Solution. During the cooling operation the enthalpy of the inside air is reduced to the value,

$$h = 25.17 + (0.01122 - 0.01033) \times 26.20 = 25.193$$

where 25.17 and 0 01033 are the values of enthalpy and humidity ratio at saturation at the apparatus dew-point temperature and 26.20 is the specific enthalpy of liquid water at that temperature. It follows that the quantity of refrigeration required is 31.514 − 25.193 = 6.321 Btu per pound of dry air.

The *inside air* being processed leaves the store with an enthalpy of 31.514 and is returned with an enthalpy of 25.17; it therefore removes energy of amount 6 344 Btu per pound of dry air. This means that the weight of dry air involved in the process is 114,510/6.344 = 18,050 lb per hour and that the total refrigeration to be supplied is 18,050 × 6.321 = 114,090 Btu per hour, or 9.508 tons.

The quantity of liquid separated out during the conditioning process is 18,050 × (0.01122 − 0.01033) = 16.018 lb per hour as required. In leaving the apparatus it takes with it energy of amount 16.018 × 26.20 = 420 Btu per hour. This plus the refrigeration accounts for the total energy removal of 114,510 Btu per hour as required.

On the Mollier Diagram, Fig. 10, the cooling operation is represented by line AB whose length is the quantity of refrigeration per pound of dry air; the separation of the liquid

formed in the cooling operation is represented by line BC whose projection on the ordinate axis is the quantity of liquid so separated per pound of dry air. Point C is the apparatus dew-point and lies on the condition line as required.

In practice it may not be feasible to choose the apparatus dew-point as the point on the condition line to which to condition the *inside air* because to do so would require an excessive number of air changes in the given space. Or it may be that the condition line does not cross the saturation curve at all so that the apparatus dew-point as defined does not exist. Finally, it is rarely possible to obtain complete saturation in conventional air conditioning apparatus. Nevertheless the requirements of the cooling load problem can be exactly met if the conditioned air is brought to *any* point on the condition line of the problem.

Heating Load

The condition line is also useful in the analysis of heating load problems as may best be illustrated by means of an illustrative example.

Example 14. A certain space is to be maintained at 70 F and 50 per cent saturation with outside conditions at 0 F and 80 per cent saturation. The normal heat transmission through walls, partitions, floor, roof, glass and doors is estimated at 75,000 Btu per hour. Energy gained from lights and appliances is estimated at 15,000 Btu per hour.

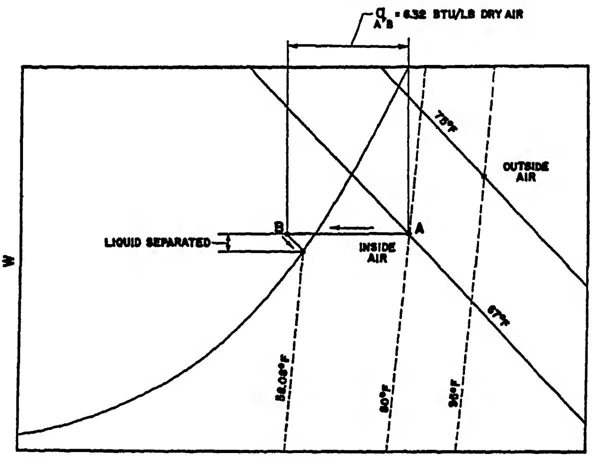

FIG. 10. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 13

Energy and water gains from occupants are to be disregarded in the calculations. Double doors and windows are used so that infiltration is negligible. The ventilation requirement is 30,000 cu ft per hour of *outside air.*

The requirements of the problem are to be met in the following manner; preheat the *ventilating air*; mix it adiabatically with recirculated *inside air*; saturate the mixture adiabatically with recirculated spray water; heat the resulting mixture to 105 F and return it to the conditioned space as *supply air.*

Analysis. Every pound of dry air admitted to the system (air conditioned space plus air conditioning apparatus) with the *ventilating air* displaces a pound of dry air from the system with *inside air.* Since the *ventilating air* is not admitted directly to the space, then for every pound of dry air withdrawn with *inside air* there is a pound of dry air returned with *supply air.* This has to have the net effect of adding energy of amount 60,000 Btu per hour and water of amount *zero* pounds per hour. Thus the ratio q determining the direction of the condition line is infinite, which means that the condition line is horizontal as indicated by the protractor on the Mollier Diagram.

The properties of *inside air* are: h = 25.451, W = 0 007910. Since the state point of the *supply air* must be on the condition line at 105 F, its properties are: h = 33.986,

$W = 0.007910$. Therefore the weight of dry air withdrawn with *inside air* and returned with *supply air* is $60{,}000 \div (33.986 - 25.451) = 7029.9$ lb per hour.

The properties of *outside air* are: $h = 0.668$, $W = 0.0006298$, $v = 11.590$. Therefore the weight of dry air introduced into the system with the *ventilating air* is $30{,}000 \div 11.590 = 2588.4$ lb per hour. This *ventilating air* is to be mixed adiabatically with *inside air* containing $7029.9 - 2588.4 = 4441.5$ lb of dry air per hour; therefore, the humidity ratio of the mixture must be $(2588.4 \times 0.0006298 + 4441.5 \times 0.007910) \div 7029.9 = 0.005229$.

The condition line crosses the saturation curve at 50.86 F where the enthalpy is 20.782 and the humidity ratio is 0.007910. This is the state point to be reached by adiabatic saturation of the mixture of *ventilating air* and *inside air* with recirculated spray water. Accordingly, the state point of the mixture must lie on the 50.86 F thermodynamic wet-bulb line so that its enthalpy must have the value,

$$h = 20.782 - (0.007910 - 0.005229) \times 18.97 = 20.731$$

This requires that the enthalpy of the preheated *ventilating air* have the value,

$$h = (7029.9 \times 20.731 - 4441.5 \times 25.451) \div 2588.4 = 12.632$$

Since the humidity ratio of the preheated *ventilating air* is known to be 0.0006298, its temperature is readily found to be 49.75 F.

The quantity of heat required for preheating the *ventilating air* is $2588.4 \times (12.632 - 0.668) = 30{,}968$ Btu per hour; that to be added to the *supply air* is $7029.9 \times (33.986 -$

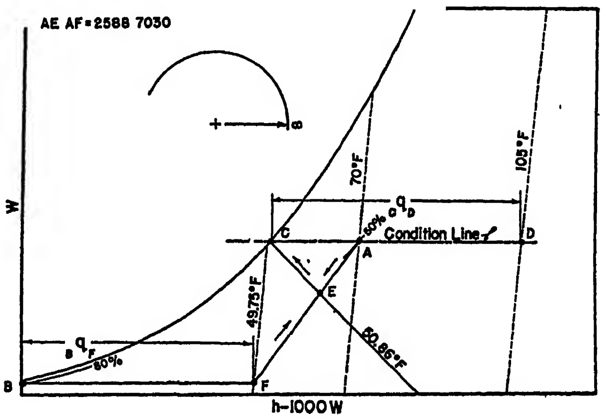

FIG. 11. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 14

$20.782) = 92{,}823$ Btu per hour; the energy added with the spray water is $7029.9 \times 18.97 \times (0.007910 - 0.005229) = 357$ Btu per hour; that introduced into the system with the *ventilating air* is $2588.4 \times 0.668 = 1729$ Btu per hour; that carried out of the system with the *inside air* displaced by the *ventilating air* is $2588.4 \times 25.451 = 65{,}877$ Btu per hour; therefore, the net energy added to the system is $30{,}968 + 92{,}823 + 357 + 1729 - 65{,}877 = 60{,}000$ Btu per hour as required.

On the Mollier Diagram, Fig. 11, point A is the state point of the *inside air*. The condition line is horizontal so that point D is the state point of the *supply air*. The condition line crosses the saturation curve at point C so that the state point of the mixture of preheated *ventilating air* and *inside air* before adiabatic saturation with recirculated spray water must lie somewhere on the thermodynamic wet-bulb line through C. The state point of the *ventilating air* is point B, hence that of the preheated *ventilating air* must lie somewhere on the horizontal line through B. Its exact location is determined graphically by finding the straight line AF which is cut by the thermodynamic wet-bulb line through C into two segments such that AE : AF = 2588.4 : 7029.9. The length of the line BF is the quantity of heat required for preheating the *ventilating air* per pound of dry air; the length of the line CD is the quantity of heat to be added to the *supply air*, per pound of dry air.

WET-BULB TEMPERATURES BELOW 32F

A condition in which the water evaporating from the wick of a wet-bulb thermometer remains liquid at 32 F or lower is one of metastable equilibrium and should therefore not be expected to occur in practice. The evidence that it does sometimes occur appears to be indirect and inconclusive. Stable equilibrium requires that the water freeze at 32 F or lower and is the condition to be expected in practice. On the Mollier Diagram the lines of constant thermodynamic wet-bulb temperature have been drawn for stable equilibrium only. In other words it has been assumed that the water evaporating from the wick of the wet-bulb thermometer freezes when its temperature falls to 32 F or lower.

Example 15. Find the temperature at which dry air has a thermodynamic wet-bulb temperature of 32 F.

Solution. If it is assumed that the water evaporating from the wick of the wet-bulb thermometer remains liquid, the specific enthalpy of the dry air must have the value,

$$h_a = 11.758 - 0.04 \times 0.003788 = 11.758$$

corresponding to which the temperature is 48.95 F. On the other hand if it is assumed that the water freezes, the specific enthalpy of the dry air must have the value,

$$h_a = 11.758 + 143.36 \times 0.003788 = 12\,301$$

corresponding to which the temperature is 51.21 F. The second assumption is the assumption of stable equilibrium and should be expected to represent the actual situation

The corresponding answer, namely 51.21 F, is the one given by the Mollier Diagram at intersection of 32 F thermodynamic wet-bulb and 0 per cent saturation.

DALTON'S RULE

As stated in the introduction the thermodynamic properties of moist air have hitherto been obtained from those of dry air and water vapor separately by application of Dalton's Rule. Actual departures from the rule are due principally, but not entirely, to intermolecular forces; therefore, in order to apply the rule with any measure of consistency it is necessary to idealize the situation by assuming that the effects of such intermolecular forces are negligible and that both the dry air and the water vapor behave like perfect gases. Making this assumption, the volume v_T occupied by n_a mols of dry air at temperature T and pressure p_a is $v_T = n_aRT/p_a$ while that occupied by n_w mols of water vapor at the same temperature but at pressure p_w is $v_T = n_wRT/p_w$. According to Dalton's Rule, if the dry air and water vapor are mixed, each occupies the whole volume of the mixture at the temperature of the mixture and the pressure of the mixture is the sum of the individual pressures. Mathematically,

$$v_T = \frac{n_aRT}{p_a} = \frac{n_wRT}{p_w} = \frac{(n_a + n_w)RT}{p} \qquad (12)$$

It follows from these equations that the so-called *partial* pressure of each constituent is its mol-fraction times the observed pressure of the mixture; thus, for water vapor,

$$p_w = \frac{n_w}{n_a + n_w}\, p \qquad (13)$$

and similarly for dry air. Equation 13 may be regarded as the Dalton Rule definition of partial pressure in terms of the observable terms n_a, n_w, p.

The humidity ratio W is the mol ratio n_w/n_a times the ratio of molecular weights, namely, 18.016/28.966 = 0.6220; hence Equation 13 can be written

$$W = 0.6220 \frac{p_w}{p - p_w} \tag{14}$$

Now, even if it is assumed that both the dry air and the water vapor behave like perfect gases, it does not follow that at saturation the *partial* pressure of the water vapor can be put equal to the saturation pressure of pure water at the temperature of the mixture because: (1) the coexisting liquid (or solid) phase is not pure water but contains a small amount of dissolved air, and (2) the coexisting liquid (or solid) phase has to support the observed pressure p and not just the saturation pressure p_s of pure water. These effects are calculable but are in general smaller than the effects of intermolecular forces which have already been ignored. Besides, the only legitimate reason for retaining Dalton's Rule is to gain simplicity; hence these effects should be disregarded also, and the humidity ratio at saturation estimated as follows,

$$W_s = 0.6220 \frac{p_s}{p - p_s} \tag{15}$$

In this chapter the ratio W/W_s has been called degree of saturation and denoted by the greek letter μ. The ratio p_w/p_s has long been called *relative humidity* and will be denoted by the Greek letter φ. Equations 14 and 15 can be combined to give

$$\mu = \varphi \frac{1 - p_s/p}{1 - \varphi p_s/p} \tag{16}$$

which can be inverted to give

$$\varphi = \frac{\mu}{1 - (1 - \mu)p_s/p} \tag{17}$$

Example 16. Find the relative humidity of moist air at 180 F, 20 per cent saturation.

Solution. Inserting numerical data from Table 1 into Equation 17, the answer is

$$\varphi = \frac{0.20}{1 - 0.80 \times 15.294/29.921} = 0.3384$$

Example 17. Find the degree of saturation of moist air at 70 F, 50 per cent relative humidity.

Solution. Inserting numerical data from Table 1 into Equation 16, the answer is

$$\mu = 0.50 \frac{1 - 0.73915/29.921}{1 - 0.50 \times 0.73915/29.921} = 0.4937$$

The foregoing examples show that there is a substantial difference between degree of saturation and relative humidity, particularly at higher temperatures. Of course, they both have the value zero for dry air and the value unity for saturated moist air regardless of the temperature.

A Dalton Rule expression for the volume of moist air per pound of dry air obtainable directly from Equation 12 is

$$v = \left(\frac{R_a T}{p}\right) + \mu \left(\frac{W_s R_w T}{p}\right) \tag{18}$$

where

R_a = gas constant for dry air = 1545.31 ÷ 28.966 = 53.349 (ft/F).

R_w = gas constant for water vapor = 1545.31 ÷ 18.016 = 85.774 (ft/F).

This expression is of the form of Equation 4.

According to Dalton's Rule the enthalpy of moist air is the sum of separate contributions from the dry air and the water vapor; thus,

$$h = h_a + \mu(W_s h_w) \tag{19}$$

where, to be consistent, the specific enthalpies h_a and h_w should be allowed to vary with temperature only, not with pressure or composition. This expression is of the form of Equation 5.

Within the accuracy of Dalton's Rule the following empirical equations give suitable values of h_a and h_w:

$$\begin{aligned} h_a &= 0.240t \\ h_w &= 0.444t + 1061 \end{aligned} \tag{20}$$

Equation 7 defining thermodynamic wet-bulb temperature may be written in the form,

$$h - h' + (W_s^* - W)h_w^* = h_s^* - h'$$

If the quantity h' that has been subtracted from both sides is understood to be the enthalpy at the thermodynamic wet-bulb temperature t^* but at the humidity ratio W, then within the accuracy of Dalton's Rule

$$\begin{aligned} h - h' &= (0.240 + 0.444W)(t - t^*) \\ h_s^* - h' &= (1061 + 0.444t^*)(W_s^* - W) \\ h_w^* &= t^* - 32 \end{aligned}$$

With these approximations Equation 7 becomes

$$W_s^* - W = \frac{0.240 + 0.444W}{1093 - 0.556t^*}(t - t^*) \tag{21}$$

Carrier [6] has modified Equation 21 by introducing further approximations as follows,

$$\begin{aligned} W_s^* &= 0.6220p_s^*/(p - p_s^*) \\ W &= 0.6220p_w/(p - p_s^*) \\ 0.444\,W &= 0 \end{aligned}$$

the first of which is part of Dalton's Rule. The result is

$$p_w = p_s^* - \frac{p - p_s^*}{2830 - 1.44t^*}(t - t^*) \tag{22}$$

except that the numerical values of the constants in the denominator of the rightmost term are somewhat different than Carrier's.

Equation 22 permits direct calculation of the partial pressure p_w from observed values of pressure p, temperature t, and wet-bulb temperature t^*, assuming that information is available regarding the saturation pressure p_s. The ratio p_w/p_s is the so-called relative humidity.

Example 18. Find the relative humidity of moist air at 90 F dry-bulb, and 63 F (thermodynamic wet-bulb).

Solution. At 63 F the value of the saturation pressure is 0.58002 in. Hg. Therefore at atmospheric pressure (29.921 in. Hg),

$$p_w = 0.58002 - 29.341 \times 27/2739 = 0.2908 \text{ in. Hg}$$

The relative humidity is

$$\varphi = 0.2908/1.4219 = 0.2045$$

the denominator being the value of saturation pressure at 90 F.

From Equation 16 may be computed the corresponding degree of saturation, the result being

$$\mu = 19.67 \text{ per cent}$$

in remarkably close agreement with the answer to Example 2.

STEADY FLOW ENERGY EQUATION

In steady flow, the energy convected by the fluid at any section is the sum of (*a*) kinetic energy due to velocity; (*b*) gravitational energy due to elevation; (*c*) enthalpy due to the condition of pressure, temperature and composition of the fluid.

Kinetic Energy

There are reasons to believe that the so-called *velocity pressure* h_v read by a Pitot tube is simply the kinetic energy per unit volume of the fluid immediately upstream from the tube, as application of Bernoulli's Equation suggests. Thus

$$V = 1097.3 \sqrt{\frac{h_v}{\rho}} \tag{23}$$

where

V = velocity, feet per minute.
h_v = velocity pressure, inches of water at 60 F.
ρ = density of fluid, pounds per cubic foot.

In the case of flow through a duct, the velocity pressure is found to vary considerably over the section and a traverse has to be made. The cross-sectional area of the duct is divided into a number of equal concentric areas, and measuring stations are located at centroidal points in each area along two perpendicular diameters. Usually the ultimate object is to determine an *average* velocity $\overline{V}$ from which the weight of fluid crossing the section per unit time can be obtained on multiplying by the cross-sectional area of the duct and by the density of the fluid. This is obtained by simply averaging the square roots of all measured velocity pressures as follows:

$$\overline{V} = \frac{1097.3}{\sqrt{\rho}} \left(h_v^{1/2}\right)_{av} \tag{24}$$

where

$\overline{V}$ = average velocity, feet per minute.
$(h_v^{1/2})_{av}$ = arithmetic average of the square roots of all measured velocity pressures, inches of water at 60 F.

But the item of present importance is the *average kinetic energy* convected with each pound of fluid. Consistently with the previous discussion, this can be shown to be

$$\overline{KE} = 0.006678\, v \frac{\left(h_v^{3/2}\right)_{av}}{\left(h_v^{1/2}\right)_{av}} \tag{25}$$

where

$\overline{KE}$ = average kinetic energy, Btu per pound.

v = specific volume, cubic feet per pound.

$(h_v^{3/2})_{av}$ = arithmetic average of the 3/2-powers of all measured velocity pressures, inches of water at 60 F.

If the velocity pressure were uniform over the section, Equations 24 and 25 could be combined to give

$$\overline{KE} = \left(\frac{\overline{V}}{13,430}\right)^2 \tag{26}$$

But, it is interesting to note that if the velocity varies *parabolically* from zero at the walls to maximum at the center as it does in the case of purely viscous flow in a circular duct, *then the average kinetic energy is twice that given by* Equation 26.

Example 19. If 2000 cfm of air flow through an 8 in. diameter circular duct, find the average kinetic energy per pound of air.

Solution. The cross-sectional area of the duct is 0.349 sq ft; hence the average flow velocity is 5730 fpm. If the velocity were uniform over the section, the average kinetic energy would be $(5730 \div 13,430)^2 = 0.182$ Btu per pound. But it is more likely that the actual distribution of velocity would approximate that characteristic of viscous flow; hence the average kinetic energy would be more nearly $2 \times 0.182 = 0.364$ Btu per pound.

Gravitational Energy

The potential energy due to elevation Z (feet) above any convenient datum is simply $Z \div 778.3$ Btu per pound of fluid. In the case of moist air,

$$\overline{PE} = \frac{\overline{Z}(1 + W)}{778.3} \tag{27}$$

where

$\overline{PE}$ = average potential energy, Btu per pound dry air.

$\overline{Z}$ = average elevation, feet.

W = humidity ratio, pound water per pound dry air.

Enthalpy

No further discussion of enthalpy is required. It may be well to emphasize, however, that enthalpies have been figured on the basis of one pound of dry air.

Heat and Shaft Work

Between any two sections 1 and 2 in an apparatus through which steady flow occurs, there may be heat *absorbed* from outside, ${}_1q_2$, Btu per pound of dry air, and shaft work *removed* to outside, ${}_1l_2$, Btu per pound of dry air. If heat is actually rejected to outside, ${}_1q_2$ is intrinsically negative; and if shaft work is actually put in from outside, ${}_1l_2$ is intrinsically negative.

Steady-flow Energy Equation

A complete energy accounting takes the form of Equation 28 which is usually referred to as the steady-flow energy equation.

$${}_1q_2 = (h_2 + \overline{KE}_2 + \overline{PE}_2) - (h_1 + \overline{KE}_1 + \overline{PE}_1) + {}_1l_2 \tag{28}$$

where

$_1q_2$ = heat *added* from outside *between* sections 1 and 2, Btu per pound dry air.
h_2 = enthalpy of the mixture *at* section 2, Btu per pound dry air.
$\overline{KE}_2$ = average kinetic energy *at* section 2, Btu per pound dry air.
$\overline{PE}_2$ = average potential energy *at* section 2, Btu per pound dry air.
h_1 = enthalpy *at* section 1, Btu per pound dry air.
$\overline{KE}_1$ = average kinetic energy *at* section 1, Btu per pound dry air.
$\overline{PE}_1$ = average potential energy *at* section 1, Btu per pound dry air.
$_1l_2$ = shaft work withdrawn *between* sections 1 and 2, Btu per pound dry air.

In Equation 28 all quantities are *per pound of dry air*. If Equation 25 is used in computing average kinetic energy, the result will be in Btu per pound of dry air if v is taken as volume per pound of dry air. If Equation 26 is used, multiplication by $(1 + W)$ as in Equation 27 is required though this is a refinement seldom justified.

Thermodynamic properties of water at saturation are given in Table 2 for the range −160 to +212 F.

U. S. STANDARD ATMOSPHERE

The so-called U. S. Standard Atmosphere is an essential standard of reference in aeronautics and as such has become important to the air conditioning engineer who frequently has to simulate atmospheric conditions at high altitudes in connection with aeronautical research. In defining this standard it is first assumed that temperature T varies linearly with altitude Z above sea level, at any rate up to the lower limit of the isothermal layer at 35,332 ft. Thus,

$$T = T_0 - 0.0019812\,Z \tag{29}$$

or

$$\frac{dT}{dZ} = -0.0019812 \text{ (degree Centigrade per foot)} \tag{30}$$

The second assumption is the validity of the perfect gas laws, namely,

$$Pv = RT \tag{31}$$

A horizontal disc of air having unit cross-sectional area (1 sq ft) and vertical thickness dZ (ft) weighs dZ/v (lb). This accounts for the difference of pressure dP (lb per sq ft) between the upper and lower faces of the disc; hence, using Equation 31

$$dZ = \frac{RT\,dP}{P} \tag{32}$$

Equations 30 and 32 can be combined to eliminate Z and then integrated to obtain the relation between pressure and temperature, namely,

$$\frac{T}{T_0} = \left(\frac{P}{P_0}\right)^{0.1903} \tag{33}$$

The values $T_0 = 288\,K$ and $P_0 = 29.921$ in. Hg are parts of the definition of the standard atmosphere.

Values of pressure and temperature are listed in Table 5 for altitudes in the standard atmosphere from −1,000 to 50,000 ft above sea level. Values for altitudes below the lower limit of the isothermal layer conform

to Equations 29 and 33. For further explanation, reference (7) should be consulted.

TABLE 5. PRESSURE AND TEMPERATURE FOR ALTITUDES IN U. S. STANDARD ATMOSPHERE

ALTITUDE FEET Z	PRESSURE IN. OF HG P	TEMP F t
− 1,000	31.02	+62.6
− 500	30.47	+60.8
0	29.921	+59.0
+ 500	29.38	+57.2
+ 1,000	28.86	+55.4
+ 5,000	24.89	+41.2
10,000	20.58	+23.4
15,000	16.88	+ 5.5
20,000	13.75	−12.3
25,000	11.10	−30.1
30,000	8.88	−47.9
35,000	7.04	−65.8
40,000	5.54	−67.0
45,000	4.36	−67.0
50,000	3.436	−67.0

LETTER SYMBOLS USED IN CHAPTER 3

μ = degree of saturation or per cent saturation.

ρ = density of fluid, pounds per cubic foot.

φ = relative humidity (decimal).

a = ratio of apparent molecular weight of dry air (28.966) to the molecular weight of water (18.016) = 1.6078.

A = coefficient from Table 4 for use in Equation 4a (obtained from Table 4).

B = coefficient to be used in Equation 5a (obtained from Table 4).

C = coefficient for use in Equation 6a (obtained from Table 4).

h = enthalpy of moist air, Btu per pound of dry air.

$\bar{h}$ = enthalpy correction term to be added above 150 F, to enthalpy.

h_a = specific enthalpy of dry air, Btu per pound.

h_{as} = $h_s - h_a$ = the difference between the enthalpy of moist air *at saturation* per pound of dry air, and the specific enthalpy of the dry air itself, Btu per pound of dry air.

h_s^* = enthalpy of moist air at saturation at *thermodynamic wet-bulb temperature*, Btu per pound of dry air.

h_s = enthalpy of moist air at saturation per pound of dry air, Btu per pound of dry air.

h_V = velocity pressure, inches of water at 60 F.

h_W = specific enthalpy of condensed water (liquid or solid) at standard pressure, Btu per pound water.

h_W^* = specific enthalpy of water as added at the *thermodynamic wet-bulb* temperature t^*, Btu per pound of dry air.

K = Kelvin degrees.

KE = kinetic energy, Btu per pound.

$\overline{KE}$ = average kinetic energy, Btu per pound.

l = shaft work withdrawn, Btu per pound of air.

${}_1l_2$ = shaft work withdrawn between sections 1 and 2, Btu per pound of air.

LETTER SYMBOLS (Continued)

m = weight of dry air crossing any duct section, pounds per minute.

m_1, m_2, m_3 = weights of dry air convected across sections F_1, F_2, F_3 respectively, pounds per minute.

M = weight of dry air withdrawn with inside air, pounds per hour.

n_a = mols of dry air.

n_w = mols of water vapor.

p = total pressure of a mixture of air and water vapor, pounds per square inch or inches Hg.

p_a = partial pressure of air, pounds per square inch or inches Hg.

p_s = *saturation* pressure of pure water vapor, pounds per square inch or inches Hg.

p_w = partial pressure of water vapor in mixture of air and water vapor, pounds per square inch or inches Hg.

P = atmospheric pressure, inches Hg.

P_0 = standard atmospheric pressure by definition 29.921 in. Hg.

PE = potential energy, Btu per pound dry air.

$\overline{\text{PE}}$ = average potential energy, Btu per pound dry air.

q = ratio of energy added (or removed) to water added (or removed), Btu per pound. Also called specific enthalpy of *water added.*

${}_1q_2$ = heat added between sections 1 and 2, Btu per pound dry air.

${}_Aq_B$ = heat added between sections A and B per pound of dry air, Btu per pound.

${}_AQ_B$ = total heat added between sections A and B, Btu per minute.

ΔQ = energy to be removed from or added to conditioned space, Btu per hour.

R = universal gas constant.

R_a = gas constant for dry air.

R_w = gas constant for water vapor.

s = entropy of moist air per pound of dry air, Btu per (pound) (Fahrenheit degree).

$\bar{s}$ = correction to be added to entropy of moist air obtained from Equation 6.

$\bar{\bar{s}}$ = additional correction to be added to entropy because of "mixing entropy" (obtained from Table 4). Correction to be added to value of s obtained from Equation 6.

s_a = specific entropy of dry air, Btu per (pound) (Fahrenheit degree, absolute).

s_{as} = the difference between the entropy of moist air *at saturation* per pound of dry air, and the specific entropy of the dry air itself, Btu per (pound of dry air) (Fahrenheit degree, absolute).

s_s = entropy of moist air at saturation per pound of dry air, Btu per (pound of dry air) (Fahrenheit degree, absolute).

s_w = specific entropy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per (pound of water) (Fahrenheit degree, absolute).

t^* = thermodynamic wet-bulb temperature, Fahrenheit degrees.

t(F) = temperature, Fahrenheit degrees.

T = absolute temperature, Fahrenheit degrees.

T_0 = standard atmospheric temperature, by definition 288 Kelvin degrees.

v = volume of moist air per pound of dry air, cubic feet per pound.

$\bar{v}$ = correction to be added to volume of moist air per pound of dry air, above 150 F.

v_a = specific volume of dry air, cubic feet per pound.

LETTER SYMBOLS (Continued)

v_{as} = $v_s - v_a$, the difference between volume of moist air *at saturation*, per pound of dry air, and the volume of the dry air itself, cubic feet per pound of dry air.

v_s = volume of moist air at saturation per pound of dry air, cubic feet per pound of dry air.

v_T = total volume, cubic feet.

V = velocity, feet per minute.

$\overline{V}$ = average velocity, feet per minute.

W = humidity ratio, of moist air, pounds of water per pound of dry air.

ΔW = water to be removed from (or added to) conditioned space, pounds per hour.

W_s = humidity ratio, *at saturation*, weight of water vapor per pound of dry air, pound per pound.

W_s^* = humidity ratio corresponding to thermodynamic wet-bulb temperature t^*, pounds of water per pound of dry air.

Z = elevation above any datum, feet.

$\overline{Z}$ = average elevation, feet.

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CHAPTER 4

Fluid Flow

Theory; Pressure Loss in Circular and Non-Circular Pipes; Compressible Fluids; Nozzles and Orifices; Steam Flow Measurement; Metering Liquids; Nozzle Coefficients and Expansion Factors; Pitot Tube; Installation of Nozzles and Orifices.

THE flow of fluids is part of the branch of engineering science known as fluid mechanics, which will be discussed here insofar as it applies to the work of engineers in the fields of heating, ventilating, and air conditioning. Probably air is the most frequently handled fluid, but other gases and liquids are often involved. Compressible fluids (gases) and incompressible fluids (liquids) vary somewhat in behavior, though in cases where pressure and density changes are small, the gases may be treated as incompressible fluids.

THEORY OF FLUID FLOW

The following head equation based on energy considerations for steady flow processes will serve as a basis for the theory of the flow of fluids. This equation is presented in several ways in various texts, but a suitable form is:

$$\frac{V_1^2}{2g} + Ju_1 + p_1v_1 + E + Jq + z_1 = \frac{V_2^2}{2g} + Ju_2 + p_2v_2 + z_2 \tag{1}$$

where

V = velocity in feet per second.
g = acceleration due to gravity = 32.17 ft per (second) (second).
J = mechanical equivalent of heat = 778 foot pounds per Btu.
u = internal energy, in Btu per pound of fluid.
p = pressure in pounds per square foot.
v = specific volume, in cubic feet per pound.
E = mechanical work, in foot pounds per pound of fluid flowing.
q = heat transferred to the fluid, in Btu per pound of fluid flowing.
z = elevation above some arbitraty datum, in feet.
Subscript 1 refers to the entrance, subscript 2 to the exit.

Introducing the enthalpy h, which by definition is $u + \frac{pv}{J}$, expressed in Btu per pound of fluid, Equation 1 becomes

$$\frac{V_1^2}{2g} + Jh_1 + E + Jq + z_1 = \frac{V_2^2}{2g} + Jh_2 + z_2 \tag{2}$$

The steady flow energy equation is applicable to a wide range of problems in the flow of fluids. Obviously it applies to flow through pipes, orifices, and nozzles, and to the flow through turbines and centrifugal pumps. Reciprocating engines and pumps are essentially intermittent, but the flow tends to become steady as the number of cylinders increases, and becomes practically uniform at the entrance and exit if the system includes receivers and pipes of sufficient size.

By substituting $\frac{p}{\rho}$ (where ρ is density in pounds per cubic foot) for

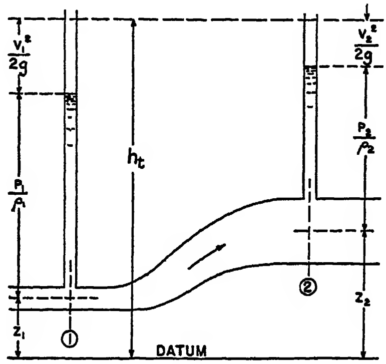

FIG. 1. RELATION OF VARIOUS FACTORS IN BERNOULLI EQUATION

its equivalent pv in Equation 1, the result, after rearranging, will be:

$$\frac{V_1^2}{2g} + \frac{p_1}{\rho_1} + z_1 = \frac{V_2^2}{2g} + \frac{p_2}{\rho_2} + z_2 + \left[J(u_2 - u_1) - E - Jq \right] \qquad (3)$$

In the case of flow through a pipe, no outside work is performed and, if the process is considered adiabatic, and if the change due to turbulence and friction is considered to be negligible, the bracketed expression in Equation 3 will disappear, leaving:

$$\frac{V_1^2}{2g} + \frac{p_1}{\rho_1} + z_1 = \frac{V_2^2}{2g} + \frac{p_2}{\rho_2} + z_2 \qquad (4)$$

which is commonly called the Bernoulli equation, named after the Swiss mathematician and physician who first propounded the theory. $\frac{V^2}{2g}$ is

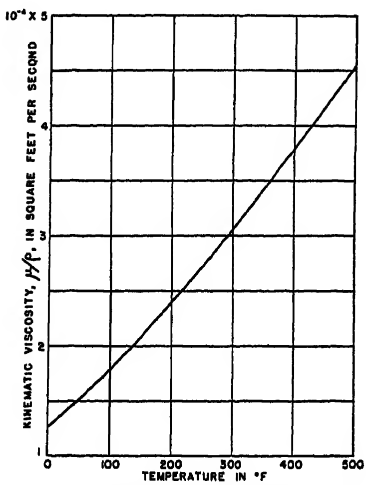

FIG. 2. RELATION OF KINEMATIC VISCOSITY TO TEMPERATURE OF AIR

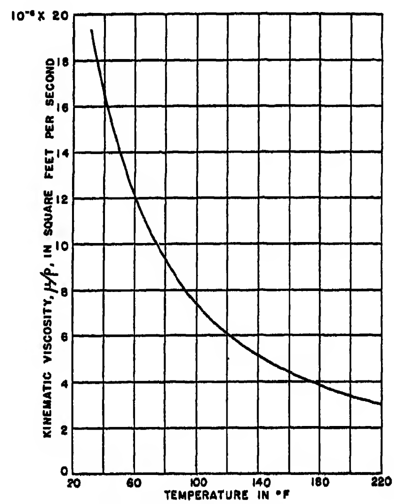

FIG. 3 RELATION OF KINEMATIC VISCOSITY TO TEMPERATURE OF WATER

known as the velocity head, $\frac{p}{\rho}$ is the pressure head, and z is the elevation head, all in feet of the fluid; the total head, h_t is the sum of the other three heads. Fig. 1 shows diagrammatically the relation of the various factors. The pressure at point 2 is lower than at point 1 because of the elevation of point 2 over point 1, and the velocity at point 2 is lower than at point 1 because of the larger pipe diameter at point 2. If the pipe diameter were the same throughout, the velocity, and consequently the velocity head, would be the same at both points, but the higher elevation at point 2 would still be responsible for a loss in pressure. The utility of the equation is evident, though it should be remembered that in it the *effects of friction and turbulence are neglected*, and that Fig. 1 represents ideal conditions. It should also be noted that the *Bernoulli equation applies only to incompressible fluids.*

Pressure Loss in Circular Pipes

The pressure loss in circular pipes is customarily expressed by the formula:

$$h_f = \frac{f\,l\,V^2}{2g\,d} \tag{5}$$

where

h_f = the loss in head of the fluid under conditions of flow, in feet.
l = the length of the pipe, in feet.
V = the velocity, in feet per second.
g = the acceleration due to gravity = 32.17 ft per (second) (second).
d = the internal diameter of the pipe, in feet.
f = a dimensionless friction coefficient.

The formula is generally known by the name of Darcy or Fanning, though it seems to have been originated by d'Aubisson de Voisins in 1834.

The factor f is a function of the Reynolds number

$$N_{Re} = \frac{d\,V\rho}{\mu} \tag{6}$$

where

N_{Re} = Reynolds number.
ρ = the density in pounds per cubic foot.
μ = the absolute viscosity in pounds per foot-second.

Both f and the Reynolds number are dimensionless. To aid in computing the Reynolds number, values of $\frac{\mu}{\rho}$, the kinematic viscosity, are shown as a function of temperature for air in Fig. 2 and for water in Fig. 3.

Fig. 4 shows the relation between f and the Reynolds number, adapted from a review by Moody[1]. The straight line sloping downward at the left of the chart supplies the values of f for laminar flow; it represents the formula

$$f = \frac{64}{N_{Re}} \tag{7}$$

With laminar flow, the velocity profile is a parabola, having the formula

[1]Superior numbers refer to the references at the end of the chapter

$$V = \frac{\rho h_f}{4\mu l}(r^2 - L^2) \qquad (8)$$

where

r = the radius of the pipe in feet.
L = distance perpendicularly from the axis of the pipe, in feet.

Accordingly, the maximum velocity occurs at the center of the pipe and is twice the average velocity; the average velocity is found when $L = 0.707\ r$. It is worth noting that roughness of the pipe wall has no effect on the loss in head for laminar flow.

Between values of the Reynolds number of 2000 and 4000, there is an

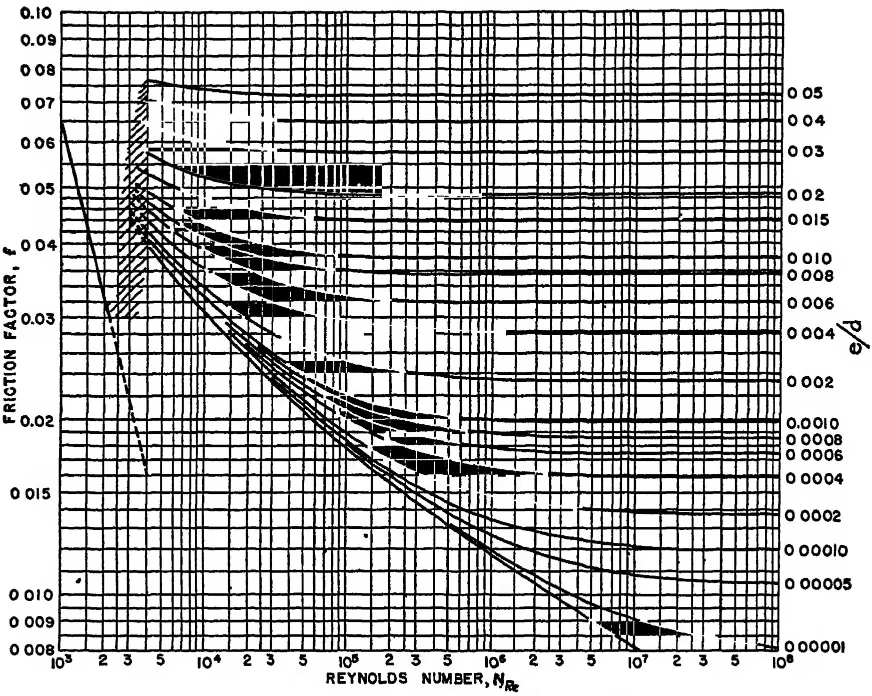

FIG. 4. RELATION BETWEEN FRICTION FACTOR AND REYNOLDS NUMBER
NOTE: The straight line at left shows values of Friction Factor for laminar flow.
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unstable region where the flow changes from laminar to turbulent, or vice versa. The actual value is impossible of prediction for any conditions of flow, though in general it may be said that the prevailing type of flow persists into the unstable region; however, once the change starts, it proceeds very rapidly.

When the flow is turbulent, the velocity profile is essentially parabolic over four fifths of the pipe diameter, but near the pipe walls, the effect of friction becomes evident, and in the boundary layer at the pipe wall the flow is laminar. Fig. 5 compares the velocity profiles for three different Reynolds numbers, but for the same average velocity.

The lower curve in the turbulent region in Fig. 4 represents the relation of f to the Reynolds number for smooth pipe, such as drawn brass tubing or glass tubing. The effect of roughness on f, which is a considerable

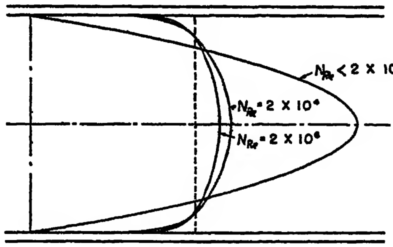

FIG. 5. COMPARISON OF VELOCITY PROFILES FOR 3 DIFFERENT REYNOLDS NUMBERS BUT FOR SAME AVERAGE VELOCITY

factor in turbulent flow, is open to some conjecture; artificially roughened pipes, for instance, give results at variance with actual tests. The curves above the smooth pipe curve of Fig. 4 represent a summary of tests on rough pipe, each of them identified by a value of e/d, with e signifying the absolute roughness in feet. Values of e/d for different pipes are given in Table 1.

To find the friction loss for any pipe, follow the curve with the proper value of e/d, to the pertinent value of N_{Re}; and from this point proceed horizontally to left margin to find the value of f to use in Equation 5.

Equation 5 is applicable to all liquids, and to gases when the pressure loss is less than 10 per cent of the initial pressure. When the loss in head is high, the formula to be used for gases is

$$\frac{p_1^2 - p_2^2}{p_1^2} = \frac{flV_1^2}{gd\, p_1 v_1} \tag{9}$$

which may be rearranged to give the loss in pressure

$$p_1 - p_2 = p_1 \left[1 - \sqrt{1 - \frac{flV_1^2}{gd\, p_1 v_1}} \right] \tag{10}$$

Pressure Loss In Non-Circular Pipes

The formulas for flow in pipes are based upon the use of pipes of circular cross-section. The formulas may be used with conduits of other shapes, and in conduits not flowing full, when the flow is turbulent, by using the hydraulic radius, R_H, which is really a ratio:

$$R_H = \frac{\text{area of cross-section}}{\text{wetted perimeter of cross-section}} \tag{11}$$

TABLE 1. VALUES OF e/d FOR DIFFERENT KINDS OF PIPE

TYPE OF PIPE	e/d
Smooth drawn tubing	0.000005
Commercial steel or wrought iron	0.00015
Asphalted cast-iron	0.0004
Galvanized iron	0.0005
Cast-iron	0.00085
Wood stave	0.0006 to 0.003
Concrete	0.001 to 0.01
Riveted steel	0.003 to 0.03

For instance, in a square duct, 1 ft on a side, handling air, the hydraulic radius is ¼ or 0.25. If the same duct is handling water, flowing 9 in. deep, the hydraulic radius is 0.75/2.5 or 0.30. Note in this latter case that the wetted perimeter does not include the distance across the free surface.

In the case of a round pipe

$$R_H = \frac{\pi d^2/4}{\pi d} = \frac{d}{4} \text{ or } d = 4R_H \tag{12}$$

Substituting Equation 12 in Equation 6,

$$N_{Re} = \frac{4R_H V\rho}{\mu} \tag{13}$$

and in the flow Equation 5,

$$h_f = \frac{flV^2}{8gR_H} \tag{14}$$

and finally in the compressible fluid flow Equation 10,

$$p_1 - p_2 = p_1\left[1 - \sqrt{1 - \frac{flV_1^2}{4g\,R_H\,p_1v_1}}\right] \tag{15}$$

Equations 13, 14, and 15 may be used to compute the flow in pipes and ducts of non-circular section and in any type of conduit not flowing full. *They should not be used when the flow is laminar.*

FLOW OF COMPRESSIBLE FLUIDS

The energy equation for the flow of compressible fluids, as represented by the gases, is derived from Equation 1. Assuming that no heat is transferred to the fluid, that no work is done, and that there is no difference in elevation, Equation 1 becomes

$$\frac{V_1^2}{2g} + Ju_1 + p_1v_1 = \frac{V_2^2}{2g} + Ju_2 + p_2v_2 \tag{16}$$

or, after rearrangement,

$$\frac{V_2^2 - V_1^2}{2g} = p_1v_1 - p_2v_2 + J(u_1 - u_2) \tag{17}$$

Since internal energy is dependent only on temperature,

$$u_1 - u_2 = c_V(T_1 - T_2) \tag{18}$$

where

c_V = the specific heat of the gas at constant volume.

T_1 and T_2 = the absolute temperatures in Fahrenheit degrees at points 1 and 2, respectively.

Substituting Equation 18 in Equation 17,

$$\frac{V_2^2 - V_1^2}{2g} = p_1v_1 - p_2v_2 + Jc_V(T_1 - T_2) \tag{19}$$

Now

$$c_V = \frac{R}{J(k-1)} \tag{20}$$

where

R = the gas constant in the expression.

$$pv = RT \tag{21}$$

k = the ratio of the specific heat at constant pressure to the specific heat at constant volume.

This ratio, k, is used extensively in fluid dynamics; values of k for various gases are given in Table 2.

TABLE 2. RATIO OF SPECIFIC HEAT AT CONSTANT PRESSURE TO SPECIFIC HEAT AT CONSTANT VOLUME FOR COMPRESSIBLE FLUIDS

COMPRESSIBLE FLUID	RATIO $k = c_p/c_v$
Helium and other monatomic gases	1.66
Air and other diatomic gases	1 40
Ammonia and hydrogen sulfide	1.34
Carbon dioxide, methane, natural gas, superheated steam, moist steam down to a quality of 97 per cent	1.28 to 1.32
Sulfur dioxide, ethylene, acetylene	1.24 to 1.26

Substituting Equations 20 and 21 in Equation 19, the energy equation becomes

$$\frac{V_2^2 - V_1^2}{2g} = \frac{k}{k-1}(p_1v_1 - p_2v_2) \tag{22}$$

While this is a convenient form of equation, it does not include all the necessary specifications. If the steady flow process is frictionless and reversible,

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^k = \left(\frac{\rho_2}{\rho_1}\right)^k \tag{23}$$

By introducing this relation in Equation 22 it is possible to reduce that equation to:

$$\frac{V_2^2 - V_1^2}{2g} = \frac{k}{k-1}\frac{p_1}{\rho_1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right] \tag{24}$$

This form of the equation is applicable not only to flow in pipes, but also to flow through orifices and nozzles.

A significant factor in the flow of compressible fluids is the velocity of sound, V_{so}, which for present purposes will be considered as the velocity at which sound will travel in the fluid at its density at the first or inlet section of the flow system being considered.

The velocity of sound is expressed as

$$V_{so} = \sqrt{\frac{gkp_1}{\rho_1}} \tag{25}$$

This formula may be developed rationally and agrees perfectly with experimental results. Substituting Equation 25 in Equation 24:

$$\frac{V_2^2 - V_1^2}{2} = \frac{V_{so}^2}{k-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right] \tag{26}$$

or, by rearrangement,

$$\frac{p_2}{p_1} = \left[1 - \frac{k-1}{V_{s0}^2}\left(\frac{V_2^2 - V_1^2}{2}\right)\right]^{\frac{k}{k-1}} \tag{27}$$

which permits the calculation of the ratio of pressures at entrance and exit of the steady flow device,—pipe, orifice, or nozzle.

FLOW THROUGH NOZZLE OR ORIFICE

Another useful expression, covering the energy change in an orifice or nozzle, may be derived from Equation 2. As with the flow through pipes, no outside work is done. Then, assuming that there is no difference in elevation, and since practically no heat is evolved or absorbed, *i.e.*, the process is adiabatic, E, z, and q of Equation 2 may be eliminated, and, by rearranging, the equation becomes

$$\frac{V_2^2}{2g} - \frac{V_1^2}{2g} = J(h_1 - h_2) \text{ foot pounds per second} \tag{28}$$

In any flow device,

$$\frac{V_1 A_1}{v_1} = \frac{V_2 A_2}{v_2} \text{ or, } V_1 = V_2 \frac{A_2 v_1}{A_1 v_2} \tag{29}$$

in which A_1 or A_2 is the cross-sectional area of the flow at a particular point, expressed in square feet. With this substituted in Equation 28, and solving for V_2:

$$V_2 = \sqrt{\frac{2gJ(h_1 - h_2)}{1 - (A_2/A_1)^2\,(v_1/v_2)^2}} \tag{30}$$

Using this expression, it is possible to determine the velocity at any point in the flow through an orifice or nozzle. If the area at the point of entry is very large with respect to that at point 2, the denominator on the right side of Equation 30 will approach unity, and the equation will reduce to

$$V_2 = \sqrt{2gJ(h_1 - h_2)} \tag{31}$$

For this reason, the expression $\sqrt{\dfrac{1}{1 - (A_2/A_1)^2\,(v_1^2/v_2^2)}}$ is called the correction factor for the velocity of approach.

The velocity of approach factor may be further simplified if the difference in volume between points 1 and 2 is negligible. Under this condition, the velocity of approach factor becomes $\sqrt{\dfrac{1}{1 - (A_2/A_1)^2}}$. If

$$\frac{A_2}{A_1} = \frac{D_2^2}{D_1^2} = \beta^2 \tag{32}$$

the velocity of approach factor is $\sqrt{\dfrac{1}{1 - \beta^4}}$, in which form it is generally used in flow formulas. The quantity β is the ratio of the throat or orifice diameter to the pipe diameter.

The connection of the velocity of sound with the flow of fluids has already been noted. Its most important application is to the flow of gases through a converging tube or nozzle. If it is assumed that the inlet velocity of the fluid, V_1, is negligible, Equation 24 will reduce to

$$V_2 = \sqrt{\left(\frac{2gk}{k-1}\right)\frac{p_1}{\rho_1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]} \tag{33}$$

Let W represent the weight of gas flowing through the converging tube in a unit of time, and A_2 the area at the throat; then,

$$W = A_2V_2/v_2 \text{ or } V_2 = Wv_2/A_2 \tag{34}$$

Substituting this, as well as the relation $p_1v_1^k = p_2v_2^k$, in Equation 33 gives,

$$W = A_2\sqrt{\left(\frac{2gk}{k-1}\right)\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]} \tag{35}$$

If this is computed and the figures are plotted, the curved line (partly solid and partly broken) of Fig. 6 is found. The maximum value of $\frac{p_2}{p_1}$ may be computed by differentiating W with respect to p_2 and equating the result to zero. This operation produces the formula:

$$\frac{p_2}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{36}$$

For air, with $k = 1.40$, $\frac{p_2}{p_1} = 0.53$.

Critical Pressure and Critical Flow

Actually, the broken part of the curve is not attained for the flow in the nozzle. If the ratio of p_2 to p_1 is decreased from unity, the weight

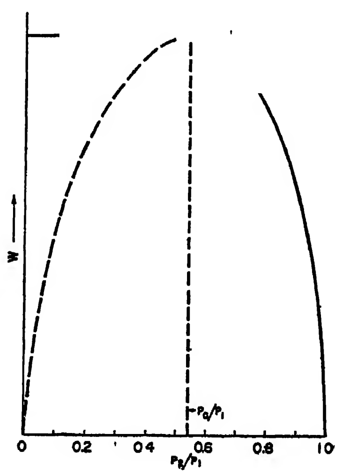

Fig. 6. Relation of Flow of Gas to Pressure Drop in a Converging Tube

rate of discharge, as well as the volume, increases from zero to a maximum, as shown by the solid section of the curve in Fig. 6; thereafter, as p_2/p_1 is decreased further, the discharge is constant, as indicated by the horizontal line. The value of p_2 at the maximum point is called the critical pressure, or p_c, and from Equation 27 it is seen that p_c is approximately 53 per cent of p_1 when air is flowing.

To find the velocity at the critical pressure, it is assumed that the upstream velocity V_1 in Equation 22 is so small as to be negligible. Using the subscript c to indicate conditions at the critical point,

$$\frac{V_c^2}{2g} = \left(\frac{k}{k-1}\right)(p_1v_1 - p_cv_c) \quad or,$$

$$V_c = \sqrt{\left(\frac{2gk}{k-1}\right)(p_1v_1 - p_cv_c)} \tag{37}$$

Substituting Equations 23 and 36, and rearranging, Equation 37 becomes

$$V_c = \sqrt{\left(\frac{2gk}{k+1}\right) p_1v_1} \tag{38}$$

and

$$V_c = \sqrt{\frac{gkp_c}{\rho_c}} \tag{39}$$

Comparing Equation 39 with Equation 25, it will be seen that the velocity at the throat is equal to the velocity of sound at the critical pressure.

Critical flow is attained only in converging tubes, in nozzles, and in orifices with a well-rounded approach. It does not occur in sharp-edged orifices or in nozzles having an expanding outlet section. The so-called critical flow prover uses this property of constant rate of flow above the critical pressure, and finds application as a flow regulator and a quantity-rate meter; in either case, the theoretical rate of flow may be computed from Equation 38, multiplying V_c by the area of the constriction to obtain the volume rate of flow.

In developing the working equations for orifices and nozzles, it is customary to start with

$$V_2^2 - V_1^2 = 2gh_f \tag{40}$$

This may be derived from the Bernoulli equation or from the relations of falling bodies. Now, since

$$A_1V_1 = A_2V_2 = Q_s \tag{41}$$

in which Q_s is the discharge rate in cubic feet per second,

$$\frac{Q_s^2}{A_2^2} - \frac{Q_s^2}{A_1^2} = 2gh_f \tag{42}$$

Transposing,

$$Q_s = \frac{A_1A_2}{\sqrt{A_1^2 - A_2^2}}\sqrt{2gh_f} \tag{43}$$

or

$$Q_s = A_2 \frac{1}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}} \sqrt{2gh_f} \tag{44}$$

The central term on the right hand side of Equation 44 will be recognized as the *velocity of approach factor*, so that

$$Q_s = A_2 \frac{1}{\sqrt{1 - \beta^4}} \sqrt{2gh_f} \tag{45}$$

Actual Flow Through Orifices and Nozzles

The actual rate of flow through an orifice, nozzle, or Venturi tube is rarely equal to the theoretical, and generally the actual rate is less than the theoretical. In the case of the nozzle and Venturi tube, this is due to losses from wall roughness, fluid friction, and turbulence during the expansion in the section following the throat. While wall roughness is not a factor in a sharp-edged orifice, fluid friction and turbulence are important, as is the fact that the discharge contracts to a degree variable with the ratio of outlet to inlet pressure after leaving the orifice, so that the limiting area is somewhat less than the opening in the orifice plate. Accordingly, Equation 45 must be modified by a correction factor, C. Usually, the velocity of approach factor is included with this correction factor, and, if

$$K = C \frac{1}{\sqrt{1 - \beta^4}}, \tag{46}$$

$$Q_s = KA_2 \sqrt{2gh_f}. \tag{47}$$

Multiplying by 3600 to convert from cubic feet per second to cubic feet per hour and converting area in square feet to diameter in inches, gives

$$Q_f = 3600\, K \frac{\pi D_2^2}{4 \times 144} \sqrt{2gh_f}$$

or

$$Q_f = 19.635\, KD_2^2 \sqrt{2gh_f} \tag{48}$$

where

Q_f = rate of flow in cubic feet per hour.
D_2 = the diameter of the orifice or nozzle throat in inches.
K = flow coefficient including correction for velocity of approach.

Equation 48 is a general equation, expressing the flow of any fluid through an orifice or nozzle. Further use of it will be made as other types of flow are discussed.

The differential loss, h_f, is in terms of feet of the fluid flowing through the orifice or nozzle. In the case of a gas flowing, where it is customary to read the differential pressure in inches of water, feet of gas must be converted to inches of water. Since dry air at 32 F and 14.7 psi absolute pressure weighs 0.0807 lb per cubic foot, the weight of a cubic foot of any other kind of gas under the same conditions is 0.0807 G, where G is the specific gravity of the gas referred to air. Water weighs 62.37 lb per cubic foot at 60 F. Using also the relation of 12 in. in 1 ft,

$$h_f = \frac{h_w}{12} \times \frac{62.37}{0.0807G} \tag{49}$$

in which h_w is the differential pressure in inches of water.

Also, since the gas flowing is not necessarily at 32 F and 14.7 psi, it is necessary to apply Charles' and Boyle's laws to the density of the gas and therefore:

$$h_f = \frac{h_w}{12} \times \frac{62.37}{0.0807G} \times \frac{14.7}{P_f} \times \frac{T_f}{492} \tag{50}$$

in which P_f and T_f are the absolute pressure and temperature of the flowing gas. Substituting this in Equation 48, and combining the constants,

$$Q_f = 218.44KD_2^2 \sqrt{\frac{h_w T_f}{P_f G}} \tag{51}$$

Then, to correct the value of Q_f to any other standard conditions of pressure P_b and temperature T_b, using the gas laws,

$$Q_b = Q_f \times \frac{P_f}{P_b} \times \frac{T_b}{T_f} \tag{52}$$

Equation 51 becomes

$$Q_b = 218.44KD_2^2 \frac{T_b}{P_b} \sqrt{\frac{P_f h_w}{T_f G}} \tag{53}$$

Finally, since gases expand under the conditions of reduced pressure downstream from the orifice or nozzle, an expansion factor, Y, must be added. The final formula, then, is

$$Q_b = 218.44KYD_2^2 \frac{T_b}{P_b} \sqrt{\frac{P_f h_w}{T_f G}} \tag{54}$$

In Equation 54, all the data must be observed at the time of measurement except K and Y. These must be obtained from charts, tables, or formulas, derived from or based on the results of a great many experiments, the results of which have been collected by a joint committee of the *American Gas Association* and the *American Society of Mechanical Engineers*[2,3]. The report of the two associations gives orifice coefficients as a function of the Reynolds number and of the ratio of orifice to pipe diameter, for pipes 2 to 12 in. and 14 in. in diameter, and for four different types of pressure taps in use in the United States. The coefficients are higher for the smaller pipe sizes. This is an effect of the turbulence produced by the roughness of the pipe surface, a given roughness being relatively greater with a small pipe than with a large one.

Space does not permit presenting all the coefficient data that are available. However, if the pipe is smooth drawn tubing, the effect of roughness is negligible, and the coefficients for the largest size of pipe apply also to smaller pipes. Figs. 7, 8, and 9 show these coefficients, N_{Re}, being the Reynolds number referred to the diameter of the orifice or throat of the nozzle, in feet.

Pressure Taps–Location and Types

The different sets of pressure taps are called flange taps, radius taps, vena contracta taps, and pipe or full-flow taps. The relative locations of the first three of these are shown in Fig. 10, and the need for different coefficients for the different taps is indicated by the course of the change in pressure of the flowing fluid shown in the lower part of the figure. Pipe taps are located 2½ pipe diameters upstream and 8 pipe diameters downstream, both measured from the upstream face of the orifice plate,

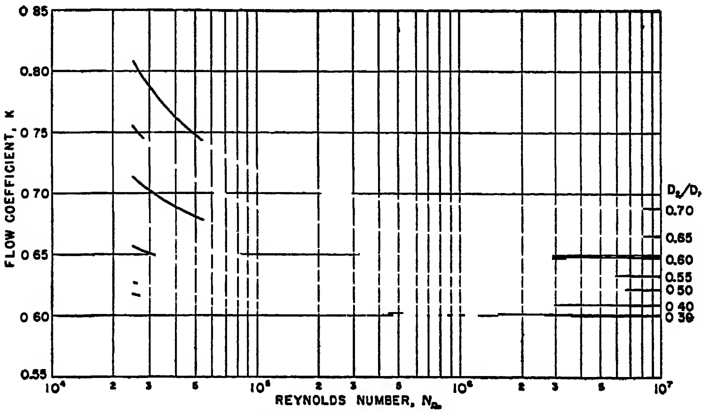

FIG. 7. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND FLANGE TAPS IN SMOOTH PIPE

NOTE: From Table 6, Bibliography [H].

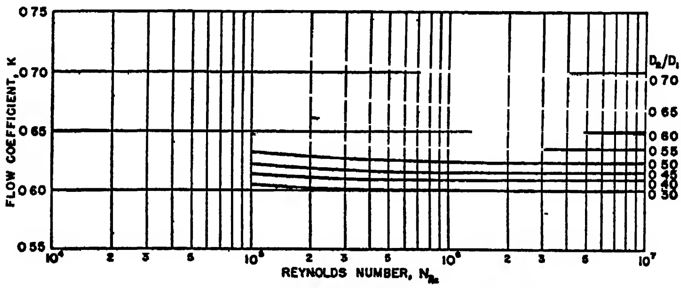

FIG. 8. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND RADIUS TAPS IN SMOOTH PIPE

NOTE: From Fig. 36d, Bibliography [K].

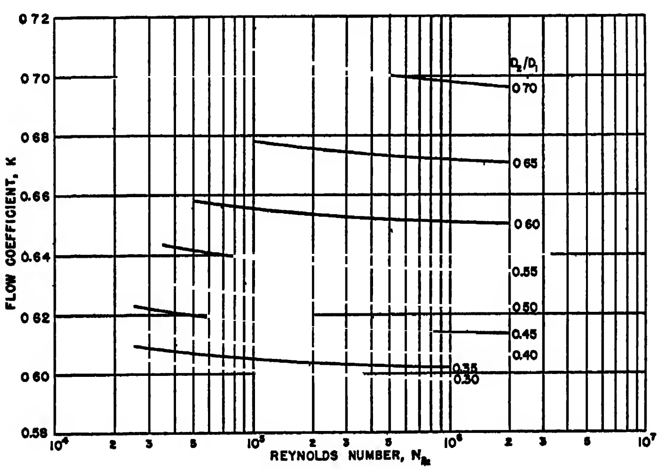

FIG. 9. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND VENA CONTRACTA TAPS, IN SMOOTH PIPE

NOTE: From Table 7, Bibliography [H].

or in other words, before the orifice plate has had any effect on the flow and after the recovery in pressure has been completed. The use of pipe or full-flow taps has been limited to the metering of natural fuel gas in certain areas. As they are not suited to use in heating and ventilating work no data for them are given in this chapter.

Still another type of pressure tap is used in European practice,—corner taps. Pressures are taken from recesses in the flange connected to annular slits in the corners formed by the pipe wall and the orifice plate. Coefficients for these taps have been adopted by the *International Standards Association*, but are not used commercially in America.

It will be noted that the location of the downstream pressure tap of the vena contracta arrangement is variable. Vena contracta is the term

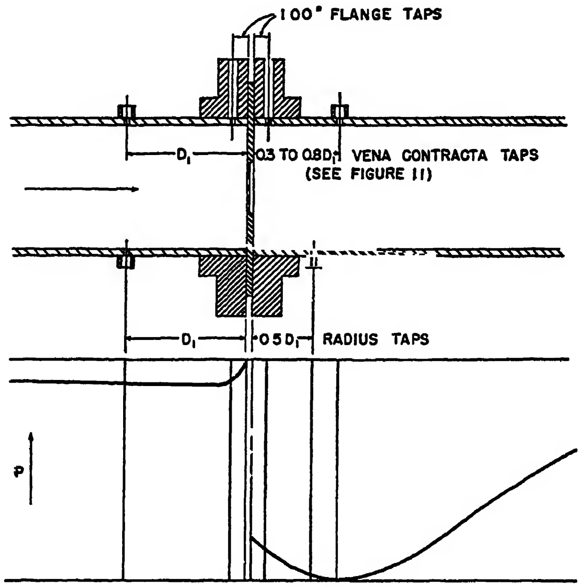

Fig. 10. Relative Location of Flange, Radius and Vena Contracta Taps

applied to the minimum cross-section of the jet from the orifice, where the static pressure is at a minimum. Its location, and the location of the downstream vena contracta tap, vary with the ratio of orifice to pipe diameter, and with rate of flow, as shown in Fig. 11; the tap is generally located in accordance with the mean curve in the figure.

Expansion Factor for Gases

The expansion factor, Y, for gases (for liquids, $Y = 1$) is found from the empirical formula

$$Y = 1 - (0.41 + 0.35\beta^4)\left(\frac{p_1 - p_2}{p_1}/k\right) \tag{55}$$

This is applicable to flange, radius, and vena contracta taps.

Values of Y for air, computed from these equations, are given in Fig. 12.

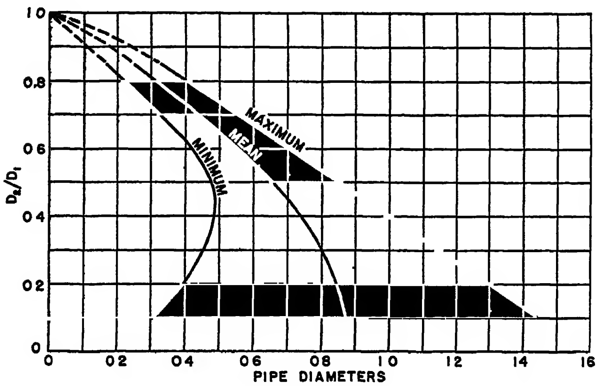

Fig. 11. Location of Vena Contracta in Relation to Ratio of Orifice to Pipe Diameter and to Rate of Flow

Computing Orifice Discharge

With this information it is possible to compute the discharge from an orifice if the Reynolds number is known. Here an odd complication is encountered—when the value of N_{Re} is computed, the rate of flow, which is the unknown quantity, must be used in the computation. However, it will be noted in Figs. 7, 8, and 9 that the orifice coefficient does not change greatly as N_{Re} changes. If, then, an estimate is made of the velocity, using this in computing N_{Re}, and if the corresponding coefficient is used in Equation 54, a value for the rate of flow will be found. Using this velocity to compute a corrected value of N_{Re} and repeating the process, a more nearly correct value of Q_f is found. This cut-and-try method may be continued for several more cycles, but generally the first or second correction will be found sufficient.

Another method would be to use the value of K corresponding to $N_{Re} = \infty$, modifying this with a factor involving the rate of flow, determined from the temperature, and the differential and static pressures. This method is used by the *American Gas Association*[8].

STEAM FLOW MEASUREMENT

While steam may be considered as a gas, its measurement differs from that of the usual gases because of a number of factors. Equation 48 serves as the starting point. Since it is usual to measure the differential pressure in inches of water, it is necessary to convert h_f, the head in feet

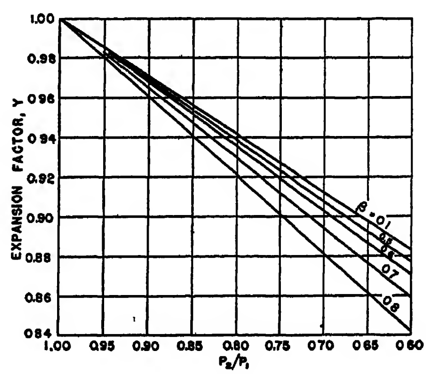

Fig. 12. Expansion Factor for Air and Other Diatomic Gases Applicable to Flange, Radius and Vena Contracta Taps

in terms of the flowing fluid, to h_{aw}, the actual head in inches of water, using the equation

$$h_f = \frac{h_{aw}}{12} \frac{\rho_w}{\rho} \tag{56}$$

where

ρ_w = the density of water at 60 F (62.37 lb per cubic foot).
ρ = the density of the flowing fluid.

Substituting Equation 56 in Equation 48 gives

$$Q_f = 359.15\, KD_2^2 \sqrt{\frac{h_{aw}}{\rho}} \tag{57}$$

In steam measurement, a constant head of water is maintained over each leg of the manometer by means of condensing chambers, in order to keep the heat of the steam away from the meter. As the mercury

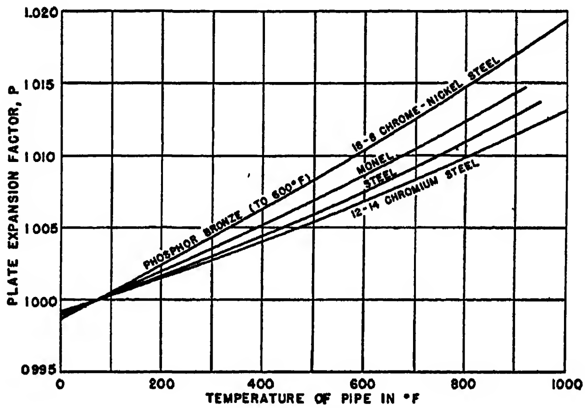

FIG. 13. VARIATION OF ORIFICE PLATE EXPANSION FACTOR, *P*, WITH TEMPERATURE AND MATERIAL

level fluctuates, the difference in head as recorded on the chart, therefore, is not that due to mercury alone, but to mercury minus an equivalent head of water. To correct for this, the equation

$$h_{aw} = h_w \frac{12.557}{13.557} \tag{58}$$

is applied to Equation 57. The denominator in Equation 58 is the specific gravity of mercury; and the numerator is the difference in specific gravity between mercury and water. Equation 57, hence, becomes

$$Q_f = 345.65\, KD_2^2 \sqrt{\frac{h_w}{\rho}} \tag{59}$$

Steam is commonly measured in terms of weight, and since

$$W = \rho Q_f \tag{60}$$

where

W = the rate of flow in pounds per hour.

$$W = 345.65\, KD_2^2 \sqrt{h_w \rho} \tag{61}$$

The expansion factor Y, and the factor P, correcting for the expansion of the orifice plate with the temperature must then be applied, so that

$$W = 345.65\, KYPD_2^2 \sqrt{h_w \rho} \tag{62}$$

which is the *final form of the equation for the flow of steam through orifices.* Values of K may be obtained from Figs. 7, 8, and 9, according to the pressure taps used. Y may be computed from Equation 55. Fig. 13 gives values of the correction factor P, according to the temperature and the material of the orifice plate. Values of the density, ρ, may be obtained from steam tables, such as Keenan and Keyes[4], which are widely used.

METERING LIQUIDS

Orifices are also used for metering liquids, and Equation 48 serves again as a starting point for developing the working formula. Again,

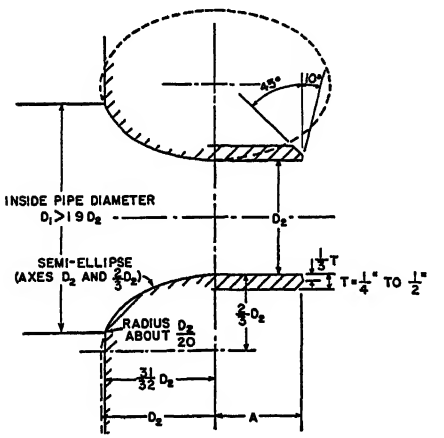

FIG. 14. SHAPE OF *ASME* LONG RADIUS NOZZLE WHEN RATIO OF THROAT TO PIPE DIAMETER IS 0.53 OR LESS

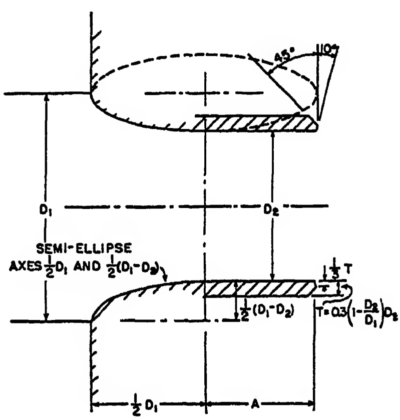

FIG. 15. SHAPE OF *ASME* LONG RADIUS NOZZLE WHEN RATIO OF THROAT TO PIPE DIAMETER IS 0 4 TO 0.7

too, it is necessary to convert h_f to h_{aw}, the actual head in inches of water, by substituting Equation 56. It is also necessary to correct for the weight of the fluid above the manometer, and since this may be other than water, it is better to use an equation of more general form than Equation 58:

$$h_{aw} = h_w \frac{13.557 - \dfrac{\rho_f}{\rho_w}}{13.557} \tag{63}$$

where

ρ_f = the density of the fluid over the mercury in the manometer.

ρ_w = the density of water at 60 F.

Substituting Equations 56 and 63 in Equation 48 gives

$$Q_f = 44\,764\, KD_2^2 \sqrt{h_w \left(\frac{1}{\rho_f} - 0.00118\right)} \tag{64}$$

Then, since liquids are generally measured in gallons instead of cubic feet, and since there are 7.4805 gal in 1 cu ft,

$$Q_W = 334.86\,KD_2^2\sqrt{h_W\left(\frac{1}{\rho_f} - 0.00118\right)} \tag{65}$$

in which Q_W is the discharge or rate of flow, in gallons per hour.

Since liquids, for practical purposes, are incompressible, no expansion factor is necessary. If circumstances demand, the factor P for the expansion of the orifice may be applied. Also, if it is necessary to correct the volumetric discharge to a base temperature, application of the known expansion characteristics of the liquid will enable the conversion to be made. Values of K again are obtainable from Figs. 7, 8, and 9, according to the type of pressure tap.

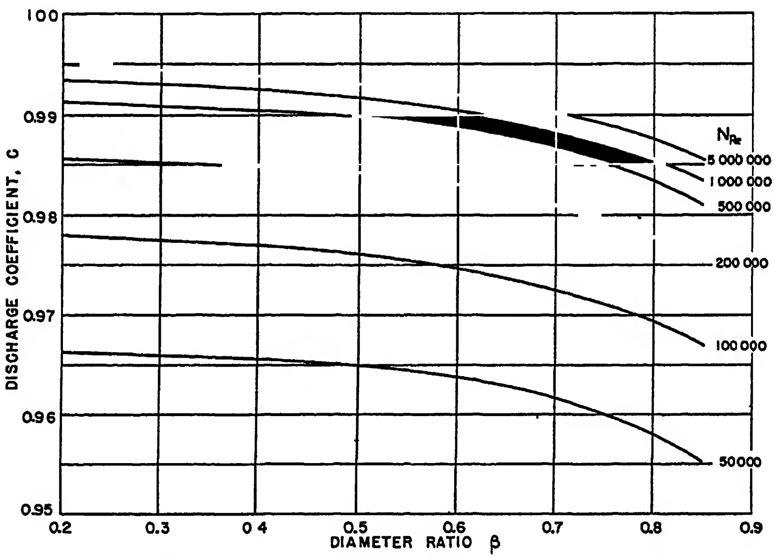

Fig. 16. Relation of Nozzle Discharge Coefficient, C, for 2-Inch Pipe, to Diameter Ratio and Reynolds Number

NOZZLE COEFFICIENTS AND EXPANSION FACTORS

Nozzles differ from orifices in that the flow is guided to the throat in such a way that contraction of the jet is suppressed, or, in other words, there is no vena contracta. Because of this fact, the coefficients are different from those of orifices, and are very close to unity before the velocity of approach factor is added. Also, the expansion factor may be deduced rationally, rather than empirically, as with orifices.

Two shapes of nozzles that have been under investigation by the *A.S.M.E.* for some time are shown in Figs. 14 and 15. They are referred to as long-radius nozzles. Their contour is that of a semi-ellipse, and the contracting portion is followed by a cylindrical section of the same area as the throat. The shape shown in Fig. 14 is designed for use with ratios of throat to pipe diameter of 0.53 or less, that of Fig. 15 for ratios of 0.4 to 0.7. The most usual location of pressure taps is 1 pipe diameter up-

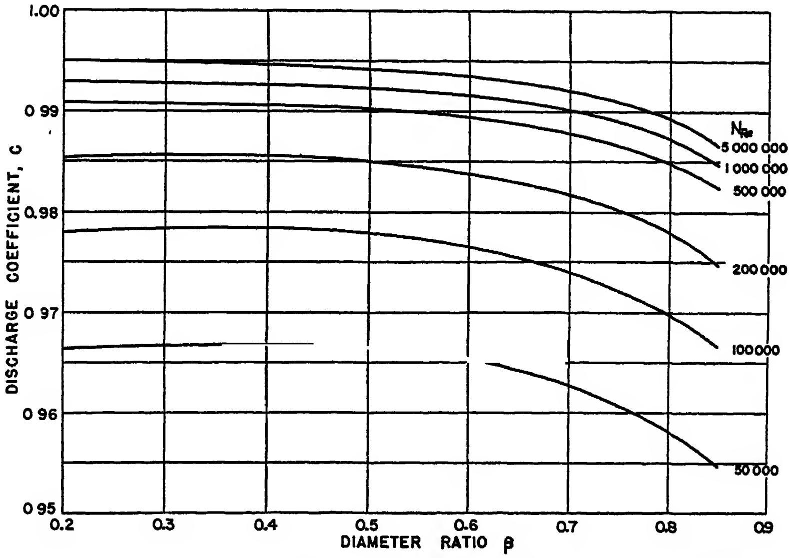

FIG. 17. RELATION OF NOZZLE DISCHARGE COEFFICIENT, *C*, FOR 6-INCH PIPE, TO DIAMETER RATIO AND REYNOLDS NUMBER

stream and ½ diameter downstream, both measured from the plane of the nozzle inlet. In addition, the *International Standards Association* has adopted still another shape of nozzle, which has a somewhat sharper approach than the *A.S.M.E.* nozzles, and which uses corner taps. Very little use of this nozzle has been made in this country.

The formulas already given for orifices apply equally to nozzles except for discharge coefficients, and for the expansion factor, when it is applied.

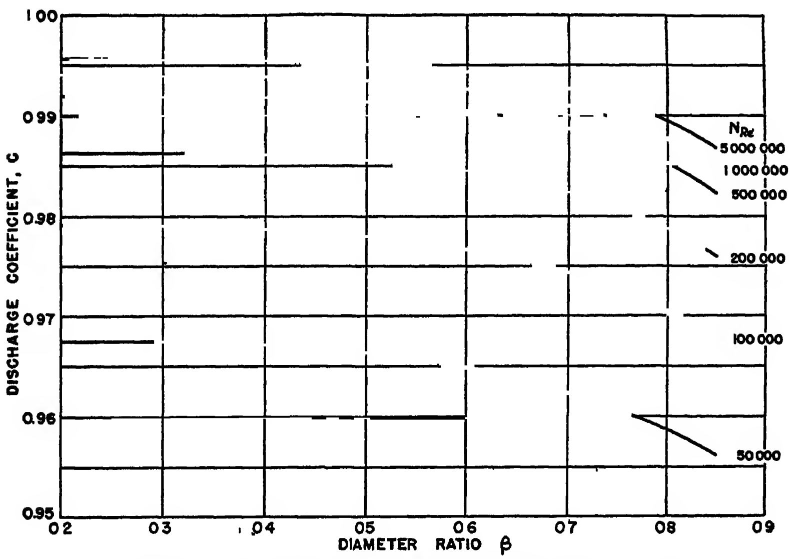

FIG. 18. RELATION OF NOZZLE DISCHARGE COEFFICIENT, *C*, FOR 10-INCH PIPE, TO DIAMETER RATIO AND REYNOLDS NUMBER

Discharge coefficients for nozzles, as for orifices, vary with pipe size; they may either increase or decrease with decreasing size of pipes depending on the sharpness of the approach curvature of the nozzle. For the *A.S.M.E.* nozzles, they tend to decrease. Generally speaking, too, the coefficient for a given nozzle shape is higher if the finish of the surface is smoother.

Discharge coefficients, C, for pipes 2, 6, and 10 in. in diameter are given in Figs. 16, 17, and 18, as correlated by Bean, Beitler and Sprenkle[5], as functions of the diameter ratio β and the Reynolds number N_{Re} (Equation 66) referred to the diameter of the throat in feet.

$$N_{Re} = \frac{D_2 V_2 \rho_1}{\mu_1} \tag{66}$$

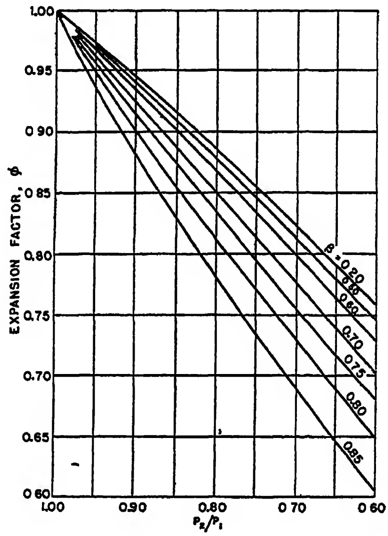

Fig. 19. Relation of Expansion Factor, φ, for Nozzles to Diameter Ratio and Pressure Loss for Air and Other Diatomic Gases

Coefficients from these curves must be multiplied by $\frac{1}{\sqrt{1 - \beta^4}}$, the velocity of approach factor, in accordance with Equation 46, to obtain the value of K to use in the various equations.

The expansion factor for nozzles, designated by φ, is obtained from a rational formula, as already noted.

$$\varphi = \sqrt{\left(\frac{p_2}{p_1}\right)^{2/k} \left(\frac{k}{k-1}\right)\left(\frac{1-(p_2/p_1)^{k\,-\,1/k}}{1 - p_2/p_1}\right)\left(\frac{1 - \beta^4}{1 - \beta^4\,(p_2/p_1)^{2/k}}\right)} \tag{67}$$

This formula is plotted for $k = 1.40$ (air and other diatomic gases) and 1.30 (steam, carbon dioxide, natural gas) in Figs. 19 and 20, respectively.

FLOW MEASUREMENT BY PITOT TUBE

There remains one other head type meter useful in ventilating work, the Pitot tube, named for the Frenchman who discovered the principle. The Pitot tube is essentially a bent tube with its open end pointed upstream, combined with another tube with its end pointed crosswise to the flow or downstream, or connected to openings crosswise to the flow. Used with flowing liquid, the liquid will rise in each tube, but higher in the one pointed upstream. Used with a flowing gas, and the two tubes connected by a U-tube containing water, the liquid level in the U-tube will be displaced, with the lower level on the side connected to the tube pointed upstream. The tube directed upstream receives the impact pressure, which is the sum of the static and kinetic pressures, while the tube directed crosswise receives only the static pressure; the difference

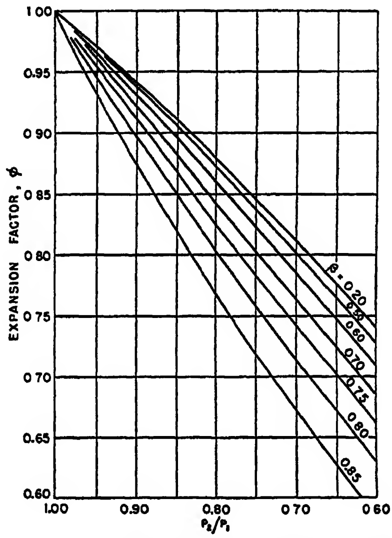

FIG. 20. RELATION OF EXPANSION FACTOR, φ, FOR NOZZLES TO DIAMETER RATIO AND PRESSURE LOSS FOR STEAM, CARBON DIOXIDE AND NATURAL GAS

between the two, as read on the separate tubes or on the U-tube is, of course, the kinetic pressure. The velocity is expressed as

$$V = \sqrt{2gh_f} \tag{68}$$

Application of Equation 56 serves to make the formula general, assuming that water is used in the manometer connecting the two tubes. Using this equation, and mutiplying by 60 to convert feet per second to feet per minute,

$$V_m = 1096.5\sqrt{\frac{h_{aw}}{\rho}} \tag{69}$$

in which V_m is the rate of flow in feet per minute.

It is often difficult to obtain the exact static pressure. In the usual construction of Pitot tubes, the static pressure openings are downstream from the impact pressure opening, and turbulence induced by the nose

may affect the static pressure reading. If the static pressure openings point downstream in any degree, a suction effect is produced to falsify the reading. In instruments having the static pressure opening pointed downstream, the coefficient may be as low as 0.86. Consequently, for accurate work, Pitot tubes should be calibrated, and the pertinent coefficient should be applied to Equation 68. In a sense, this coefficient is advantageous, since it results in a higher differential reading, which, in turn, enables more accurate readings at low flows.

In using Pitot tubes, it is generally necessary to make a traverse of the pipe or duct to determine the course of the velocity pattern. In a pipe, for instance, one of the profiles shown in Fig. 5 would be obtained. Near a valve or fitting, however, the profile might be quite distorted, a fact which would be revealed by the traverse. If the pipe or duct is

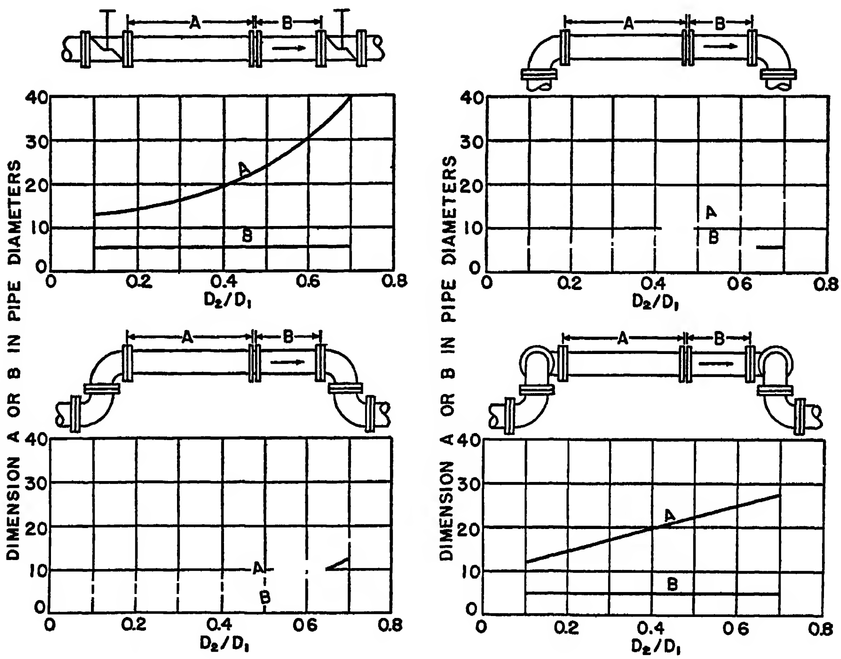

FIG. 21. MINIMUM CONDITIONS TO BE OBSERVED WHEN INSTALLING ORIFICES AND NOZZLES BETWEEN FITTINGS AND VALVES

divided into equal areas, and a determination of h_{aw} is made for each, the average velocity would be obtained by using the average of the square roots of h_{aw} in Equation 69.

INSTALLATION OF NOZZLES AND ORIFICES

A final note should be made of the installation of orifices and nozzles. Generally speaking, the orifice or nozzle, together with a holding arrangement, including pressure taps, is available from the manufacturer. In making the installation, the user should make certain that the flow approaching the nozzle or orifice is steady and evenly distributed, *i.e.*, with velocity profiles similar to those shown in Fig. 5. Fittings and valves,

which tend to direct the flow to one side and which in some cases cause it to rotate as it advances, must be far enough upstream from the orifice or nozzle to permit the disturbed stream to straighten out to the even form before reaching the meter. Minimum conditions in the installation for avoiding trouble from fittings and valves are shown in Fig. 21. If necessary, straightening vanes may be used upstream from the orifice or nozzle at a distance of not less than 8 pipe diameters.

LETTER SYMBOLS USED IN CHAPTER 4

β = ratio, throat or orifice diameter to pipe diameter.
μ = absolute viscosity in pounds per foot second.
μ/ρ = kinematic viscosity in square feet per second.
ρ = density of flowing fluid in pounds per cubic foot.
ρ_w = density of water at 60 F, (62.37 lb per cubic foot).
ρ_f = density of fluid over mercury in a manometer.
φ = expansion factor for nozzles.
A = cross-sectional area of flow, in square feet.
C = correction factor (coefficient of discharge) for flow through orifice, nozzle or Venturi.
c_p = specific heat of gas at constant pressure.
c_v = specific heat of gas at constant volume.
D = diameter of fluid stream in feet.
d = internal diameter of pipe in feet.
E = mechanical work in foot pounds per pound of fluid flowing.
e = absolute roughness of pipe surface, in feet.
f = dimensionless friction coefficient.
g = acceleration due to gravity = 32.17 ft per (second) (second).
G = specific gravity of gas referred to air.
h = enthalpy, Btu per pound of fluid.
h_{aw} = loss of head in inches of water.
h_f = loss of head in feet of fluid.
h_t = total head in feet of fluid.
h_w = differential pressure in inches of water.
J = mechanical equivalent of heat = 778 foot pounds per Btu
K = flow coefficient (correction factor), including velocity of approach correction factor, for flow through orifice, nozzle or Venturi.
k = ratio of specific heat at constant pressure to specific heat at constant volume
L = perpendicular distance from axis of pipe in feet.
l = length of pipe in feet.
N_{Re} = Reynolds number.
P = correction factor for expansion of orifice plate with temperature.
p = pressure in pounds per square foot.
p_c = critical pressure.
P_b = standard pressure to which correction is to be made, pounds per square inch, absolute.
P_f = pressure of gas flowing, pounds per square inch, absolute.
Q_b = rate of flow in cubic feet per hour under standard conditions of pressure and temperature.
Q_f = rate of flow in cubic feet per hour.
Q_s = discharge rate in cubic feet per second.
Q_w = rate of flow in gallons per hour.
q = heat transferred to the fluid per pound of fluid flowing.
R = gas constant.
R_H = hydraulic radius = ratio of area of cross-section to wetted perimeter of cross-section.
r = radius of pipe in feet.

T = temperature, Fahrenheit degrees, absolute.
T_b = standard temperature to which correction is to be made, Fahrenheit degrees absolute.
T_f = temperature of gas flowing, Fahrenheit degrees, absolute.
u = internal energy, Btu per pound of fluid.
V = velocity in feet per second.
V_c = critical velocity, feet per second.
V_{so} = velocity of sound, feet per second.
V_m = velocity in feet per minute.
v = specific volume, cubic feet per pound.
W = weight of gas flowing, pounds per hour.
Y = expansion factor—correcting for expansion of gas under reduced downstream pressure.
z = elevation above some arbitrary datum, in feet.

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CHAPTER 5

Fundamentals of Heat Transfer

Conduction, Convection, Radiation, Combined Convection and Radiation, Heat-Flow Resistance, Practical Heat Transfer Problems, Unit Conductances for Convection Flow Systems, Radiation Factors or Emissivities, Solutions for Steady-State Conduction Problems

HEAT is that form of energy that is transferred by virtue of an existing temperature difference. The temperature difference is the potential which causes the transfer, the latter in turn being resisted by the thermal properties of the material combined in a single term known as the resistance. Energy exchange associated with evaporation, condensation, etc. is treated elsewhere such as in the section on *cooling tower design* in Chapter 37. The objectives of this chapter are to:

1. Describe the mechanisms and present the rate equations for the different modes of heat transfer.

2. Illustrate the application of the basic concepts to steady-state problems (temperature independent of time or a cyclic variable thereof) by means of several typical solutions of heat transfer systems.

Further applications to specific systems will be found throughout THE GUIDE.

CONDUCTION, CONVECTION AND RADIATION

Thermal conduction is the term applied to the mechanism of heat transfer whereby the molecules of higher kinetic energy transmit part of their energy to adjacent molecules of lower kinetic energy by direct molecular action. Since the temperature is proportional to the average kinetic energy of the molecules, thermal transfer will occur in the direction of decreasing temperature. The motion of the molecules is random; there is no net material flow associated with the conduction mechanism. In the case of flowing fluids, thermal conduction is significant in the region very close to a solid boundary or wall, for in this region the flow is laminar, parallel with the wall surface, and there are practically no cross currents in the direction of the heat transfer across the solid fluid boundary. In solid bodies the significant mechanism of heat transfer is always thermal conduction.

Contrasted to the thermal conduction mechanism, *thermal convection* involves energy transfer by eddy mixing and diffusion [1] in addition to conduction. This is shown schematically in Fig. 1 which exhibits transfer from a pipe wall at surface temperature t_s to a colder fluid at a bulk temperature t_f. (Bulk temperature is that which would be attained if the fluid stream were drawn off at a certain section and mixed. It is therefore slightly higher than the lowest temperature in the stream). In the *laminar* sublayer, immediately adjacent to the wall, the heat transfer occurs by thermal conduction; in the *transition* region, which is called the buffer layer, eddy mixing as well as conduction effects are significant; in the *eddy* or *turbulent* region the major fraction of the transfer occurs by eddy mixing.

In most commercial equipment the main body of the fluid is in turbulent flow, and the laminar film exists at the solid walls only, as shown in

Fig. 1. But in cases of low-velocity flow in small tubes, or with viscous liquids such as heavy oil (low Reynolds numbers), the entire flow may be laminar. In these latter cases there is no transition or eddy region.

When the fluid currents are produced by sources external to the heat transfer region, as for example by a pump, the described solid to fluid heat transfer is termed *forced convection.* In contrast, if the fluid currents are generated internally, as a result of non-homogeneous densities arising from the temperature variations, the heat transfer is termed *free convection.*

In the conduction and convection mechanisms heat is transferred as internal energy, *i.e.*, the random molecular kinetic energy associated with the material temperature. For *radiant heat transfer*, however, a change in energy form takes place from internal energy at the source to electromagnetic energy for transmission, then back to internal energy at the receiver.

The rate of heat transfer, corresponding to the three transfer mechanisms previously described, may be expressed by three rate equations.

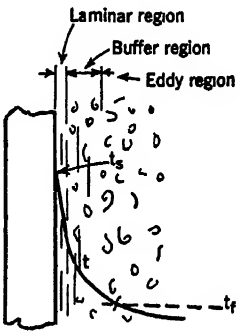


Fig. 1. Thermal Convection Conditions

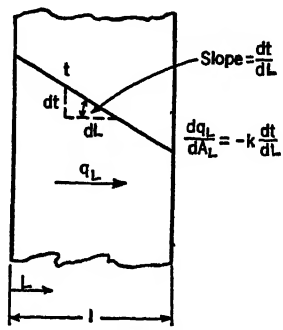


Fig. 2. Thermal Conduction in a Flat Slab

These are similar to Ohm's Law for electrical flow, the current flow through a resistance being proportional to the potential difference.

Thermal Conduction Equation

Equation 1 states symbolically that the thermal conduction per unit transfer area normal to the flow, $(dq)/(dA)$, Btu per (hour) (square foot), is proportional to the temperature gradient $(dt)/(dL)$, Fahrenheit degrees per foot. The proportionality factor is termed the *thermal conductivity*, k, Btu per (hour) (square foot) (Fahrenheit degree per foot of thickness).

$$\frac{dq}{dA} = -k\frac{dt}{dL} \tag{1}$$

The minus sign on the right side of the equation is introduced to indicate positive transfer in the direction of decreasing temperature. Fig. 2 shows the physical significance of indicated quantities.

It should be emphasized that the thermal conductivity used should be expressed in consistent units; either using the inch or foot throughout.

Expressions of conductivity used in the heating field are usually

inconsistent in this sense, in that it is customary to refer to the conductivity *per square foot* but for one inch of thickness. This custom has been adopted for the reason that wall thicknesses are usually expressed in inches, whereas if expressed in feet, decimal or fractional thicknesses would result. When dealing with flat walls no complication is involved in using the inconsistent expression of conductivity. However, when curved or spherical walls are considered, considerable complication is involved. Therefore, in this discussion the consistent units of conductivity expressed in *Btu per (hour) (square foot) (Fahrenheit degrees per one foot thickness)* are used throughout. *Conductivity values obtained from Chapter 6 or Table 1 in this chapter, which are expressed in inconsistent units, must therefore be converted for use in the calculations of this chapter by dividing by 12.* As an example, the conductivity of brick, expressed in inconsistent units as 5.0 in Table 2 of Chapter 6, becomes 0.42 when

TABLE 1. APPROXIMATE UNIT THERMAL CONDUCTIVITIES[a]

Conductivity, k = Btu per (hr) (sq ft) (F deg per in.)

MATERIAL	k	MATERIAL	k
Air	0.168	Lead	240.0
Aluminum	1416.0	Nickel	408.0
Brass (70 − 30)	720.0	Soil	2.4—12.0
Cast-Iron	336.0	Steel, mild	312.0
Copper	2640.0	Water, liquid	4.08
Glass	3.6—7.32		

[a]Thermal conductivities depend to some extent on temperature. The above magnitudes are approximate only Refer to Heat Transmission, 2nd edition, by W. H McAdams (McGraw-Hill Co., 1942) for additional values.

used in the calculations of this chapter. Also, it should be emphasized that in order to make the calculations and applications consistent in this chapter, *all dimensions of thickness must be expressed in feet.*

Thermal Convection Equation

$$\frac{dq}{dA} = h_c (t_s - t_f) \tag{2}$$

This rate equation states that the thermal convection per unit transfer area $(dq)/(dA)$, Btu per (hour) (square foot) is proportional to the temperature difference, $(t_s - t_f)$ which is the temperature of the surface less that of the fluid. The particular fluid temperature to use for a given system will be noted under the discussion of that system. The proportionality factor is termed the *unit convection conductance* (sometimes called the film coefficient for convection), h_c, Btu per (hour) (square foot) (Fahrenheit degree). These convection conditions are illustrated in Fig. 1.

The heat transmission by free or natural convection for objects surrounded by air can be conveniently expressed as in Equation 2a:

$$q_c = C \left(\frac{1}{D}\right)^{0.2} \left(\frac{1}{T_{av}}\right)^{0.181} (t_s - t_f)^{1.27} \tag{2a}$$

where

q_c = heat transmission by convection, Btu per (square foot) (hour).

C = a constant depending upon the shape of the surface.

D = diameter of pipe or circular duct or height of vertical wall, inches. (Effect of diameter or height becomes constant at 24 in.)

T_{av} = average of wall surface and surrounding air temperature, Fahrenheit degrees absolute.

$t_s - t_f$ = temperature excess between wall surface and surrounding air, Fahrenheit degrees.

For horizontal cylinders, the value of $C = 1.016$ has been well established by various investigations. For vertical plates, the value of $C = 1.394$ has been fairly well established. Suggested values [2] of C for horizontal plates warmer than the surrounding air are 1.79 when facing upward and 0.89 when facing downward.

TABLE 2. HEAT TRANSMISSION BY FREE CONVECTION FOR LARGE VERTICAL SURFACES

Expressed in Btu per (square foot) (hour)

	Temperature Difference between Body and Surrounding Still Air at 80 F													
F Deg	0	10	20	30	40	50	60	70	80	90	100	110	120	130
0	0	4.4	10.4	17.4	25.0	33.2	41.8	50.6	59.9	69.4	79.4	89.2	99.4	109.8
1	0.3	4.9	11.1	18.1	25.8	34.1	42.6	51.5	60.8	70.3	80.4	90.2	100.4	110.9
2	0.6	5.5	11.8	18.9	26.7	34.9	43.5	52.4	61.8	71.3	81.4	91.2	101.5	112.0
3	1.0	6.0	12.5	19.7	27.5	35.7	44.3	53.4	62.7	72.3	82.4	92.2	102.6	113.0
4	1.4	6.6	13.2	20.5	28.3	36.6	45.2	54.3	63.7	73.3	83.3	93.3	103.6	114.1
5	1.8	7.3	13.9	21.2	29.2	37.4	46.1	55.2	64.6	74.3	84.2	94.3	104.7	115.2
6	2.3	7.9	14.6	22.0	30.0	38.3	47.0	56.1	65.6	75.3	85.2	95.3	105.7	116.3
7	2.8	8.5	15.3	22.7	30.8	39.1	47.8	57.1	66.5	76.3	86.2	96.3	106.7	117.3
8	3.3	9.1	16.0	23.5	31.6	40.0	48.7	58.0	67.5	77.4	87.2	97.4	107.8	118.4
9	3.8	9.7	16.7	24.3	32.4	40.9	49.7	59.0	68.4	78.4	88.2	98.4	108.8	119.5

The heat transmission by free convection from vertical walls *24 in. or more in height* is given in Table 2 as calculated from Equation 2a for an ambient air temperature of 80 F. The values in Table 2 are not changed appreciably by a considerable change in air temperature for a given temperature excess. For instance, a change in air temperature from 80 to 40 F will increase the heat transmission given in Table 2 by only 1.3 per cent.

Table 2 can also be used for calculating the free convection rate of transmission for various commercial shapes such as pipes and ducts. These calculations are simplified by the use of the factors in Tables 3 and 4. Table 3 gives factors by which the values in Table 2 must be multiplied to obtain the free convective transfer from various shapes *whose characteristic dimensions are 24 in. or over*, and Table 4 gives the factors to be used in conjunction with the factors in Table 3 for obtaining the free convection from Table 2 for pipes and ducts *whose characteristic dimensions are less than 24 in.*

For example, the free convection transfer from a 3 in. O.D. horizontal cylinder for a temperature difference of 40 deg = 25.0 × 0.73 × 1.52 = 27.7 Btu per (square foot) (hour).

Problems in either forced convection or natural convection may be solved by the simple first-power equation if the convection coefficient is expressed as a unit conductance:

$$q_c = h_c A\ (t_1 - t_2) \qquad (2b)$$

where

q_c = heat transmission by convection, Btu per hour.

A = surface area, square feet.

$t_1 - t_2$ = temperature difference between the surface and the fluid Fahrenheit degrees.

h_c = unit conductance, from Table 5, Btu per (square foot) (hour) (Fahrenheit degree temperature difference.)

TABLE 3. FREE CONVECTION FACTORS FOR VARIOUS SHAPES

SHAPES	FACTOR
Horizontal cylinders 24 in. in diam. or over	0.73
Long vertical cylinders 24 in. in diam. or over	0.88
Vertical plates 24 in. in height or over	1.00
Horizontal plates warmer than air facing upward	1.28
Horizontal plates warmer than air facing downward	0.64
Horizontal plates cooler than air facing upward	0.64
Horizontal plates cooler than air facing downward	1.28

TABLE 4. FREE CONVECTION FACTORS FOR VARIOUS DIAMETER PIPES OR VARIOUS HEIGHT PLATES

Actual O.D., or height, in	1	2	3	4	5	6	7	8
Factor	1.88	1.64	1.52	1.43	1.37	1.32	1.28	1.25
Actual O.D., or height, in	9	10	12	14	16	18	20	22
Factor	1.22	1.19	1.15	1.11	1.09	1.06	1.04	1.02

Thermal Radiation Equation

The relation shown in Equation 3 is usually applicable to systems in which radiant exchange takes place between the surfaces of solids, as sche-

$$q_r = \sigma A_1 F_A F_E (T_1^4 - T_2^4) \qquad (3)$$

matically shown in Fig. 3. Gaseous and luminous radiation are not considered in this discussion. Equation 3 states that the net radiation per unit transfer area of surface 1, q_r/A Btu per (hour) (square foot), which *sees* surface 2 through a non-absorbing medium, is proportional to the difference of the fourth powers of the absolute surface temperatures $(T_1^4 - T_2^4)$. The proportionality factor (σ $F_A F_E$) may be conveniently separated into three parts:

σ = the Stefan-Boltzmann radiation constant.

= 1730×10^{-12} Btu per (hour) (square foot) (Fahrenheit degree absolute temperature to the fourth power).

F_A = the configuration factor which is dimensionless and $\leqq 1$. This factor accounts for the shape and relative position of the two surfaces. The value of $F_A = 1$ may be used in the cases of large parallel planes, long concentric cylinders or smaller bodies in large enclosures.

F_E = the emissivity factor which is also dimensionless and $\leqq 1$. This factor accounts for the absorption and emission characteristics of the surfaces for the radiation which exists. Individual emissivities (ε) should be taken from Table 6 and applied, for either radiation or absorption, as follows:

a. For a small body in a large enclosure, use the emissivity of the small body only: $F_E = \varepsilon_1$.

b. For large parallel planes, long concentric cylinders or large enclosed bodies, use both emissivities in the equation:

$$F_E = \frac{1}{\frac{1}{e_1} + \frac{1}{e_2} - 1}$$

The radiation under black-body conditions, or for an emissivity of 1.0, is given in Table 7[8] for cold surfaces as low as −39 F to warmer surfaces as high as 139 F. The emissivities of a number of surfaces ordinarily encountered in engineering practice are shown in Table 6. For radiation table at higher temperatures, and further discussion of radiation calculations, see Chapter 31.

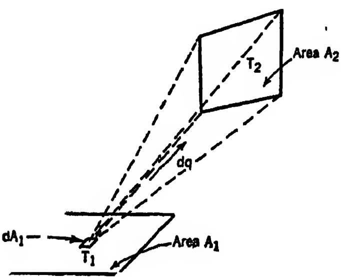


FIG. 3. RADIATION BETWEEN SURFACES

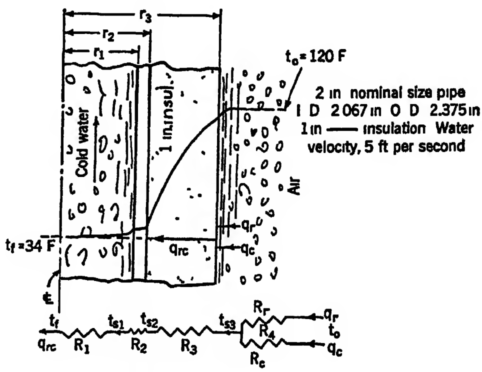


FIG. 4. HEAT TRANSFER CONDITIONS IN THE INSULATED COLD WATER LINE

NOMENCLATURE AND DIMENSIONS FOR TABLE 5

c_p = fluid unit heat capacity at constant pressure, Btu per (pound) (Fahrenheit degree).

D = cylinder diameter, feet.

G = 3600 $V_s \rho$ = fluid mass velocity, pounds per (hour) (square foot of flow cross-section).

ρ = density, pounds per cubic foot.

h_c = unit conductance for thermal convection, Btu per (hour) (square foot) (Fahrenheit degree).

k = unit thermal conductivity of the fluid, Btu per (hour) (square foot) (Fahrenheit degree per one foot thickness).

R_H = hydraulic radius of the flow cross-section = flow cross-section area per wetted perimeter, feet.

s = fin spacing, feet.

t = average fluid film temperature, Fahrenheit degree.

$t_1 - t_2$ = temperature difference surface to main fluid, Fahrenheit degree.

V_s = fluid velocity, feet per second.

μ = fluid viscosity, pounds per (hour) (foot) = viscosity in centipoises × 2.42.

Combined Convection and Radiation

It should be noted that the previous equations and tables give the heat transfer by convection and by radiation computed separately. In many

TABLE 5. APPROXIMATE UNIT CONDUCTANCES FOR THERMAL CONVECTION FOR SEVERAL FLOW SYSTEMS[a]

Expressed in Convenient Empirical Form

CASE	SYSTEM	UNIT CONDUCTANCE EQUATION[b]
	FORCED CONVECTION	
1.	Longitudinal flow in cylinders, turbulent region. Fluid being heated[c].	$\frac{h_c D}{k} = 0.0225 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c p \mu}{k}\right)^{0.4}$ For $\left(\frac{DG}{\mu}\right) > 3000$
2.	For longitudinal air flow in cylinders case 1 reduces to[c].	$h_c = 0.0036\, G^{0.8}/D^{0.2}$ For $\left(\frac{DG}{\mu}\right) > 3000$
3.	For longitudinal water flow in cylinders case 1 reduces to[c].	$h_c = 0.00486\,(1 + 0.01t)\,\frac{G^{0.8}}{D^{0.2}}$ For $\left(\frac{DG}{\mu}\right) > 3000$
4.	Air flow normal to a single right circular cylinder.	$h_c = 0.45 \left(\frac{k}{D}\right) + 0.178\, G^{0.56} \left(\frac{k}{D}\right)^{0.44}$
5.	Air flow over staggered pipe banks.	$h_c = 0.061 \left(\frac{k}{D}\right)^{0.31} G^{0.69}$
6.	Air flow over single spheres.	$h_c = 0.040\,\frac{G^{0.52}}{D^{0.48}}$ $0 < t < 250$ F
7.	Air flow over plane surfaces	$h_c = 1 + 0.22\, V_s$ For $V_s < 16$ fps or $h_c = 0.53\, V_s^{0.8}$ $16 \text{ fps} < V_s < 100$ fps
8.	Air flow normal to finned cylinders.	$h_c = 6.2 \left(\frac{G}{3600}\right)^{0.8} \frac{s^{0.32}}{D^{0.52}}$ $0 < t < 250$ F
	FREE CONVECTION[d]	
9.	Single horizontal right circular cylinder in air.	$h_c = 0.23 \left(\frac{t_1 - t_2}{D}\right)^{0.25}$
10.	Vertical surfaces in air.	$h_c = 0.3\,(t_1 - t_2)^{0.25}$
11.	Top surface of horizontal plates to air.	$h_c = 0.4\,(t_1 - t_2)^{0.25}$
12.	Bottom surface of horizontal plates to air	$h_c = 0.2\,(t_1 - t_2)^{0.25}$

[a]Heat Transmission, by W. H. McAdams.

[b]Fluid properties should be evaluated at the arithmetic mean fluid temperature, $t_f = (t_{\text{surface}} + t_{\text{fluid}})$ divided by 2.

[c]These expressions are applicable to longitudinal flow in other than right circular cylinders provided the hydraulic radius is employed as the conduit dimension parameter For non-circular cross-sections $D = 4\,R_H$.

[d]For low rates of heat transfer by free convection the exponent decreases towards zero, and for higher rates increases towards 0.33. The following equations employing an exponent equal to 0 25 are applicable in the intermediate range.

practical cases it is desirable to treat convection and radiation as a single combined process, using a first-power equation:

$$q_{rc} = h_{rc} A (t_1 - t_2) \tag{4}$$

where q_{rc} is the total heat flow due to radiation and convection, in Btu per hour. Values of h_{rc}, the surface or film conductance for combined

TABLE 6. RADIATION FACTORS OR EMISSIVITIES, ε.
For the determination of factor F_E in Equation 3

CLASS	SURFACES	FRACTION OF BLACK-BODY RADIATION		ABSORPTIVITY FOR SOLAR RADIATION
		At 50–100 F	At 1000 F	
1	A small hole in a large box, sphere, furnace, or enclosure	0.97 to 0.99	0.97 to 0.99	0.97 to 0.99
2	Black non-metallic surfaces such as asphalt, carbon, slate, paint, paper	0 90 to 0.98	0.90 to 0.98	0.85 to 0.98
3	Red brick and tile, concrete and stone, rusty steel and iron, dark paints (red, brown, green, etc.)	0.85 to 0.95	0.75 to 0.90	0.65 to 0.80
4	Yellow and buff brick and stone, firebrick, fire clay	0.85 to 0.95	0.70 to 0.85	0.50 to 0.70
5	White or light-cream brick, tile, paint or paper, plaster, whitewash	0.85 to 0.95	0 60 to 0.75	0.3 to 0.5
6	Window glass	0.90 to 0.95	....	Transparent[a]
7	Bright aluminum paint; gilt or bronze paint	0.4 to 0.6	....	0 3 to 0 5
8	Dull brass, copper, or aluminum; galvanized steel; polished iron	0.2 to 0.3	0 3 to 0 5	0.4 to 0.65
9	Polished brass, copper, monel metal	0.02 to 0 05	0.05 to 0.15	0.3 to 0.5
10	Highly polished aluminum, tin plate, nickel, chromium	0 02 to 0.04	0.05 to 0.10	0.10 to 0.40

[a]Reflects about 8 per cent

radiation and convection, are given in Chapter 6, Table 1 and Fig. 3. Complete tables for the combined heat transfer of steam and hot water radiators, pipes, coverings, etc., will be found in the appropriate chapters.

When dealing with the effect of operating temperatures upon the combined heat transfer of a given piece of equipment (as for instance a steam radiator), another form of equation is frequently used:

$$q_{rc} = B A (t_1 - t_2)^n \tag{5}$$

Values of n in this equation usually range from 1.3 to 1.5 (see Chapter 25). The chief advantage of this equation is the convenience of representing heat transfer performance on logarithmic coordinates, and the factor B should be regarded as a simple constant of proportionality.

HEAT-FLOW RESISTANCE

In most of the steady-state heat transfer problems encountered in air conditioning applications, more than one of the heat transfer mechanisms are effective, and the thermal current flows through several resistances in series or in parallel. In using the resistance concept the calculations involved are analogous to the application of Ohm's Law in electricity, *viz.*, the heat flow or thermal current is directly proportional to the thermal potential or temperature difference, and inversely proportional to the thermal resistance:

TABLE 7. HEAT TRANSMISSION BY RADIATION FOR BLACK-BODY CONDITIONS[a]

Expressed in Btu per (square foot) (hour)

TEMP F DEG	0	−1	−2	−3	−4	−5	−6	−7	−8	−9
−30	59.3	58.7	58.2	57.7	57.2	56.7	56.2	55.7	55.2	54.7
−20	65.2	64.7	64.1	63.5	62.9	62.3	61.7	61.1	60.5	59.9
−10	71.4	70.8	70.1	69.5	68.9	68.3	67.7	67.1	66.4	65.8
0	78.0	77.4	76.7	76.0	75.4	74.7	74.0	73.4	72.7	72.1
	0	+1	+2	+3	+4	+5	+6	+7	+8	+9
0	78.0	78.7	79.4	80.1	80.8	81.5	82.2	82.9	83.6	84.3
10	85.0	85.7	86.5	97.2	88.0	88.7	89.4	90.2	90.9	91.7
20	92.4	93.3	94.0	94.8	95.6	96.4	97.2	98.0	98.8	99.6
30	100	101	102	103	104	105	105	106	107	108
40	109	110	111	112	112	113	114	115	116	117
50	118	119	120	121	122	123	123	124	125	126
60	127	128	129	130	131	132	133	134	135	136
70	137	138	139	140	142	143	144	145	146	147
80	148	149	150	151	152	153	154	155	156	157
90	159	160	161	162	163	164	166	167	168	169
100	170	171	173	174	175	176	178	179	180	182
110	183	184	185	187	188	189	191	192	193	195
120	196	197	199	200	201	203	204	206	207	209
130	211	212	214	215	217	218	220	221	222	224

[a]*Example:* Radiation from walls of room at 32 F to surface at − 25 F for effective emissivity of 0.95 = (102 − 62 3) 0.95 = 37.7 Btu per (square foot) (hour).

$$q_{rc} = \frac{t_1 - t_2}{R} \tag{6}$$

Following the electrical analogy, when there is a thermal current flowing through several *resistances in series*, the resistances are additive:

$$R_T = R_1 + R_2 + R_3 + \ldots\ldots + R_n \tag{7}$$

Similarly, conductance is the reciprocal of resistance, and for heat flow through several *resistances in parallel*, the conductances are additive:

$$C_T = \frac{1}{R_T} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \ldots\ldots + \frac{1}{R_n} \tag{8}$$

Practical Heat Transfer Problems

The use of these relations for resistance and conductance makes possible the solution of many practical heat transfer problems. As discussed in Chapters 6, 7 and 28, the practical analyses of heat transfer in building walls, in fin-tube coils and in pipe coverings, are usually computed by this

method. The same resistance analysis may be applied to complicated steady-state *conduction* problems. Table 8 gives the resistances in six common cases of steady-state conduction.

A complete analysis by the resistance method is well illustrated by considering the heat transfer from the air outside to the cold water inside of an insulated pipe. The temperature gradients and the nature of the resistance analysis are indicated by the two sketches of Fig. 4.

Since air is sensibly transparent to radiation, there will be some heat transfer by both radiation and convection to the outer insulation surface. The mechanisms act in parallel on the air side. The total transfer by radiation and convection then passes through the insulating layer and the pipe wall by thermal conduction, and thence by convection and radiation into main cold water streams. (Radiation is not significant on the water side as liquids are sensibly opaque to radiation, although water transmits energy in the visible region). The contact resistance between the insulation and the pipe wall is presumed to be equal to zero.

Referring to Fig. 4, the heat transferred for a given length N of pipe, q_{rc}, Btu per hour, may be thought of as flowing through the parallel resistances R_r and R_c, associated with the insulation surface radiation and convection transfer. Then the flow is through the resistance offered to thermal conduction by the insulation, R_3, through the pipe wall resistance, R_2, and into the water stream through the convection resistance, R_1. Note the analogy to the direct current electrical circuit problem. A temperature (potential) drop is required to overcome these resistances to the flow of thermal current. The total resistance to heat transfer, R_T, hour Fahrenheit degrees per Btu, is the summation of the individual resistances:

$$R_T = R_1 + R_2 + R_3 + R_4 \qquad (9)$$

where the resultant parallel resistance R_4 is obtained from:

$$\frac{1}{R_4} = \frac{1}{R_r} + \frac{1}{R_c}$$

Provided the individual resistances may be evaluated, the total resistance can be obtained from this relation. Then the heat transfer for the length of pipe (N, ft) can be established by the relation:

$$q_{rc} \text{ (Btu per hour)} = \frac{t_o - t_f}{R_T} \qquad (10)$$

For a unit length of the pipe the heat transfer rate is:

$$\frac{q_{rc}}{N} \text{ (Btu per hour foot)} = \frac{(t_o - t_f)}{R_T N} \qquad (11)$$

The temperature drop, Δt, through an individual resistance may then be calculated from the relation:

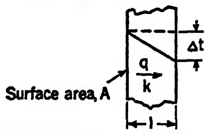
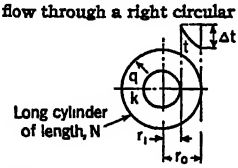
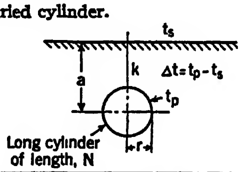
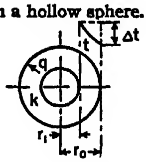
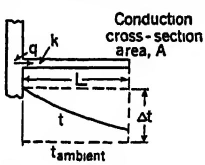
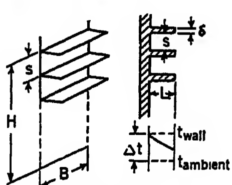
$$\Delta t = R\, q_{rc}$$

where R is the resistance in question.

The problem is now reduced to one of evaluating the individual resistances of the system. This entails suitable integration of the rate Equations 1, 2 and 3 to produce expressions of the form:

$$q = \frac{\Delta t}{R} \qquad (12)$$

TABLE 8. SOLUTIONS FOR SOME STEADY-STATE THERMAL CONDUCTION PROBLEMS[a,b]

No.	SYSTEM	Expressions for the resistance R entering into the equation: $q = \Delta t/R$ (Btu per hour)
1.	Flat wall or curved wall if curvature is small (wall thickness less than 0.1 of inside diameter). 	$R = \dfrac{L}{kA}$
2.	Radial flow through a right circular cylinder. 	$R = \dfrac{\log_e \dfrac{r_0}{r_1}}{2\pi kN}$ (See footnote *c*).
3.	The buried cylinder. 	$R = \dfrac{\log_e \dfrac{2a}{r}}{2\pi kN}$; $R = \dfrac{\cosh^{-1} \dfrac{a}{r}}{2\pi kN}$ for $\dfrac{a}{r} \geq 3$ (See footnote *c*).
4.	Radial flow in a hollow sphere. 	$R = \dfrac{\dfrac{1}{r_1} - \dfrac{1}{r_0}}{4\pi k}$
5.	The straight fin or rod heated at one end. 	$R = \dfrac{m}{h_s p \tanh mL}$ (see footnotes *d* and *e*). For $ml > 2.3$, $\tanh m L \approx 1$ $m = \sqrt{h_s p/kA}$ A = conduction cross-section area. p = perimeter of cross-section A. h_s = unit conductance to the surroundings from the fin surface. k = thermal conductivity fin material. Δt = wall temperature—ambient temperature
6.	Finned surface of area HB.  Surface area, HB	$R = \dfrac{(s + \delta)}{h_s \left(\dfrac{2}{m} \tanh m\, l + s\right) HB}$ $m = \sqrt{\dfrac{h_s p}{kA}} = \sqrt{\dfrac{2\, h_s}{k\delta}}$ Δt defined as in Case 5 above.

[a]The dimensions to be employed in these solutions are: length of dimension p, L, r = feet; units of k = Btu per (hour) (square foot) (Fahrenheit degree for one foot thickness); units of h, Btu per (hour) (square foot) (Fahrenheit degree); units of area, A = square feet.

[b]The thermal conductivity, k, in these solutions should be taken at the average material temperature (see Table 5).

[c]$\log_e x = 2.303 \log_{10} x$.

[d]This expression can also be employed as an approximation for tapered fins or of annular fins by employing average magnitudes of A and p.

[e]Tanh is the hyperbolic tangent.

where q is the heat transfer rate, and Δt is the potential drop or temperature difference through the resistance R. Table 8 lists such solutions for six different conduction systems. Table 2 in Chapter 6 and Table 1 of this chapter indicate the magnitudes of the thermal conductivities, k, to be employed in the expressions of Table 8, after dividing k by 12.

The solution applicable to the problem depicted in Fig. 4, for the calculation of R_2 and R_3, is case 2 in Table 8. Thus for a *1 ft length* of 2 in. nominal size pipe (I. D. = 2.067 in., O. D. = 2.375 in.) insulated with 1 in. of material having a conductivity of 0.025:

$$R_2 = \frac{\log_e \frac{1.188}{1.033}}{2\pi \times 26 \times 1} = 8.5 \times 10^{-4} \text{ hr Fahrenheit degree per Btu.}$$

$$R_3 = \frac{\log_e \frac{2.188}{1.188}}{2\pi \times 0.025 \times 1} = 3.9 \text{ hr Fahrenheit degree per Btu.}$$

The convection resistances to heat transfer from the pipe wall to the cold water, R_1, and from the air to the surface of the insulating material, R_c, are dependent on the flow conditions prevailing at these surfaces, and on the thermal properties of the fluids. The unit conductances for thermal convection, h_c, Btu per (hour) (square foot) (Fahrenheit degree), have been determined by test for many flow systems. These data may be employed to predict the conductances for *similar flow systems*. Table 5 summarizes some empirical equations expressing such test results.

For the problem under consideration (Fig. 4) case 3 of Table 5 is applicable for the calculation of the cold water side convection resistance R_1. Corresponding to the water velocity of 5 fps, the mass velocity is:

G = 5 (ft per sec) $\times$ 62.4 (lb per cu ft) $\times$ 3600 (sec per hr) = 11.2 $\times$ 10^5 lb per (hour) (square foot).

The inside diameter of the pipe D is 2.067/12 = 0.1725 ft.

The average water film temperature will be estimated as 36 F (mixed mean fluid temperature of 34 F). Then case 3, Table 5 yields:

$$h_c = 0.00486\,(1 + 0\,36)\,\frac{(11.2 \times 10^5)^{0.8}}{(0.1725)^{0.2}} = 650 \text{ Btu per (hr) (sq ft) (F deg).}$$

The transfer area on which this conductance is based is the inside tube area. Associated with 1 ft length of pipe there are:

$$\pi \times \frac{2.067}{12} \times 1 = 0.542 \text{ sq ft.}$$

Thus the resistance for 1 ft of tube length is:

$$R_1 = \frac{1}{h_c \pi D \times 1} = \frac{1}{650 \times 0.542} = 2.8 \times 10^{-3} \text{ hr Fahrenheit degree per Btu.}$$

Case 9, Table 5 is applicable for calculating the free thermal convection resistance, R_c, existing between the surrounding air and the insulation. The air temperature is given as 120 F. As an approximation a 20 deg temperature difference between the air and the pipe surface will be assumed. D = 4.375/12 = 0.364 ft. Then case 9 yields:

$$h_c = 0.23 \left(\frac{20}{0.364}\right)^{0.25} = 0.63 \text{ Btu per (hour) (square foot) (Fahrenheit degree).} \quad (13)$$

This result may not be deemed conservative inasmuch as the expression is for *still* air. If, however, the air is not still, but flows at approximately 5 mph or 7 fps the mass velocity corresponds to:

$$G = 7 \times 0.07 \times 3600 = 1770 \text{ lb air per (hour) (square foot).}$$

A magnitude of $k = 0.014$ Btu per (hour) (square foot) (Fahrenheit degree) per one foot thickness applied to case 4 yields:

$$h_c = 0.45 \left(\frac{0.014}{0.364}\right) + 0.178\,(1770)^{0.56} \left(\frac{0.014}{0.364}\right)^{0.44}$$

$$= 0.017 + 2.8 = 2.8 \text{ Btu per (hour) (square foot) (Fahrenheit degree).}$$

This conductance is based on 1 sq ft of outside lagging area. Thus, since there are $\pi \times (4.375/12) = 1.14$ sq ft of outside lagging area associated with 1 ft length of pipe:

$$R_c = \frac{1}{2.8 \times 1.14} = 0.312 \text{ hr Fahrenheit degree per Btu.}$$

The radiation resistance, R_r, which acts in parallel with the convection resistance, R_c, for the transfer of heat to the surface of the insulation, may be calculated. For the purposes of this illustrative problem it will be assumed that the insulated pipe is exposed to (*sees*) surroundings, which exist at 120 F. Then the angle factor, F_A, is unity and for an estimated surface emissivity of 0.9 (see Table 6), $F_e = 0.9$. As a first approximation the insulation surface temperature will be estimated as 20 deg below the surroundings at 120 F. Then the radiation per degree of temperature difference, by Equation 3 (or more conveniently by Table 8) divided by the temperature difference will be:

$$h_r = \frac{(196 - 170)\,0.95}{20} = 1.17 \text{ Btu per (hour) (square foot) (Fahrenheit degree).}$$

The outside surface area of the insulation associated with 1 ft of pipe length was previously calculated as 1.14 sq ft. Thus:

$$R_r = \frac{1}{1.17 \times 1.14} = 0.75 \text{ hr Fahrenheit degree per Btu.}$$

The resultant resistance of R_c and R_r acting in parallel (see Fig. 4) can now be evaluated as:

$$\frac{1}{R_4} = \frac{1}{R_c} + \frac{1}{R_r} = \frac{1}{0.312} + \frac{1}{0.75} = 4.54 \text{ Btu per (hour) (Fahrenheit degree).}$$

$$R_4 = 0.22 \text{ hr Fahrenheit degree per Btu.}$$

The over-all resistance, R_T, surroundings to cold water, is the sum of $R_1 + R_2 + R_3 + R_4 = 4.1$ hr F deg per Btu for 1 ft length of pipe. Note that the controlling resistances are R_3 and R_4 and that neglect of both R_1 and R_2 would not significantly influence the total resistance, R_T.

On the basis of this resistance calculation the heat transfer from the surroundings to the cold water may be evaluated as:

$$\frac{q_{rc}}{N} = \frac{t_0 - t_f}{R_T} = \frac{120 - 34}{4.1} = 21 \text{ Btu per (hour) (foot)}$$

or about 0.175 tons of refrigeration per 100 ft of pipe.

Since the calculation is based on a 1 ft pipe length:

$$q_{rc} = 21 \text{ Btu per hour.}$$

The temperature drops through the various resistances are now readily evaluated by Equation 12 as:

$t_o - t_{s3}$ air to insulation surface $= R_4\, q_{rc} = 0.22 \times 21 = 4.6$ F.
$t_{s3} - t_{s2}$ through the insulation $= R_3\, q_{rc} = 3.9 \times 21 = 82$ F.
$t_{s2} - t_{s1}$ through the pipe wall $= R_2\, q_{rc} = 8.5 \times 10^{-4} \times 21 = 0.02$ F.
$t_{s1} - t_f$ pipe wall to cold water $= R_1\, q_{rc} = 2.8 \times 10^{-3} \times 21 = 0.06$ F.

The solution was obtained on the assumption that the air temperature and the outside temperature differed by 20 deg. In order to obtain a slightly better estimate of the rate of heat transfer the numerical solution should be repeated using the temperatures calculated from the previous listed temperature differences.

The foregoing problem serves to illustrate a general method of solving steady-state heat transfer problems. There are many problems which cannot be approximated by steady-state solutions. For instance, the problem of pipe line insulation in transient service; the behavior of automatically controlled thermoflow circuits; or the periodic absorption of solar energy by roof and wall structures during the day and nocturnal radiation to the *cold* sky at night. The transient heat transfer problem differs from the steady-state in that energy storage rates need to be considered. Thus thermal capacity in addition to resistance effects is significant. The vector sum of the thermal capacitance and resistance is the thermal impedance. It is not within the scope of this chapter to deal with these problems. There are, however, solutions available in graphical form for certain special cases. Also a general approximate method may be employed which is analogous to the treatment of capacity-resistance lumped parameter electrical circuits.

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[1]—Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co., 1937).

[2]—The Transmission of Heat by Radiation and Convection, by Griffith and Davis (*Special Report* No. 9, 1922, *Department of Scientific and Industrial Research*, His Majesty's Stationery Office, London, England).

[3]—Heat Insulation in Air Conditioning, by R. H. Heilman (*Industrial and Engineering Chemistry*, Vol. 28, July, 1936, p 782).

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Industrial Heat Transfer, by Alfred Schack (John Wiley & Sons, 1933).

Calculation of Heat Transmission, by M. Fishenden and O. A. Saunders (His Majesty's Stationery Office, London, 1932).

CHAPTER 6

Heat Transmission Coefficients of Building Materials

Heat Transfer Symbols; Calculating Over-all Coefficients; Conductivity of Homogeneous Materials; Surface Conductance; Air Space Conductance; Practical Coefficients and Their Use; Computed Heat Transmission Coefficients; Roof Coefficients; Combined Ceiling and Roof Coefficients; Basement Floor, Basement Wall, and Concrete Slab Floor Coefficients, Condensation in Buildings

THE design of conditioning or heating systems for buildings requires a knowledge of the thermal properties of the walls enclosing the space. The rate of heat flow through the walls under steady-state conditions at design temperatures is usually the basis for calculating the heat required. For a given wall under standard conditions the rate is a specific value designated as U, the *over-all coefficient of heat transmission.* It may be determined by test in a guarded hot box apparatus or it may be computed from known values of the thermal conductance of the various components. Because testing of all combinations of building materials is impracticable, the procedure and necessary data for calculation of the value of U are given in this chapter, together with tables of computed values for the more common constructions.

HEAT TRANSFER SYMBOLS

U = over-all coefficient of heat transmission (air to air); the time rate of heat flow expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference between air on the inside and air on the outside of a wall, floor, roof or ceiling). The term is applied to the usual combinations of materials in construction and also to single materials, such as window glass, and includes the surface conductance on both sides.

k = thermal conductivity; the time rate of heat flow through a homogeneous material under steady conditions through unit area per unit temperature gradient in the direction perpendicular to the area. Its value is expressed in Btu per (hour) (square foot) (Fahrenheit degree per inch). Materials are considered homogeneous when the value of k is not affected by variation in thickness or size of sample within the range normally used in construction.

C = thermal conductance; the time rate of heat flow through a material from one of its surfaces to the other per unit temperature difference *between the two surfaces.* Its value is expressed in Btu per (hour) (square foot) (Fahrenheit degree). The term is applied to specific materials as used which may be either homogeneous or heterogeneous.

f = film or surface conductance; the time rate of heat flow between a surface and the surrounding air. Its value is expressed in Btu per (hour) (square foot of surface) (Fahrenheit degree temperature difference). Subscripts i and o are used to differentiate between inside and outside surface conductances respectively.

a = thermal conductance of an air space; the time rate of heat flow through an air space per unit temperature difference *between the boundary surfaces.* Its value is expressed in Btu per (hour)(square foot of area)(Fahrenheit degree). The conductance of an air space is dependent on the temperature difference, the height, the depth, the position and the character of the boundary surfaces. The relationships are not linear and accurate values must be obtained by test and not by computation.

R = thermal resistance. Its value is expressed in Fahrenheit degrees per (Btu) (hour) (square foot). It may represent any of the following and must therefore be properly described:

$$\frac{1}{U} = \text{over-all or air-to-air resistance}$$

$$\frac{1}{k} = \text{resistance per unit thickness (resistivity)}$$

$\frac{1}{C}$ = resistance of a material (surface-to-surface)

$\frac{1}{f}$ = film or surface resistance

$\frac{1}{a}$ = air space resistance

CALCULATING OVER-ALL COEFFICIENTS

From Chapter 5, Equation 7, the total resistance to heat flow through a wall is equal numerically to the sum of the resistances in series. Then by definition,

$$U = \frac{1}{R_t} = \frac{1}{R_1 + R_2 + R_3 + \cdots + R_n} \quad (1)$$

where

R_1, R_2, etc. are the individual resistances of the wall components.

R_t = total resistance.

For a wall of a single homogeneous material of conductivity k and thickness x, with surface coefficients f_i and f_o

$$U = \frac{1}{R_t} = \frac{1}{\frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o}} \quad (2)$$

For a compound wall of three homogeneous materials in series, having conductivities k_1, k_2 and k_3 and thicknesses x_1, x_2 and x_3 respectively, and laid together without air spaces,

$$U = \frac{1}{R_t} = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o}} \quad (3)$$

For a wall with air space construction and consisting of two homogeneous materials of conductivities k_1 and k_2, thicknesses x_1 and x_2, and separated by an air space of conductance a

$$U = \frac{1}{R_t} = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o}} \quad (4)$$

In the case of types of building materials having non-uniform or irregular sections such as hollow clay tile or concrete blocks, it is necessary to use the conductance C of the section unit as manufactured instead of a conductivity k. The resistance of the section $\frac{1}{C}$ is therefore substituted for $\frac{x}{k}$ in Equations 2, 3 and 4.

CONDUCTIVITIES AND CONDUCTANCES

The method of calculating the over-all coefficient of heat transmission for a given construction is comparatively simple, but accurate values of conductivities and conductances must be used to obtain satisfactory results. In addition there are sometimes parallel heat flow paths of different resistances in the same wall, which require modification of the

formula. In such cases calculated results should be checked by test measurements.

The determination of the fundamental conductivities and conductances requires considerable skill and experience to obtain accurate results. It is recommended that thermal conductivities of homogeneous materials be determined by means of the Guarded Hot Plate [1]. For determination of conductances, a Guarded Hot Box method [2] is generally used.

Tables 1 and 2 give conductivities and conductances which are quite generally used in calculation and which have been selected from various sources. Wherever possible the properties of the material and test conditions are given. In selecting and applying heat transmission values to any construction, caution is necessary, because coefficients for the same material may differ because of variations which occur in test methods, in the materials themselves, or in the temperature of the material when tested.

Conductivity of Homogeneous Materials

Thermal conductivity is a property of a homogeneous material and of types of building materials such as lumber, brick and stone which may be considered homogeneous. Most insulating materials, except air spaces and reflective types, are of a porous nature and consist of combinations of solid matter with small air cells. The thermal conductivity of these materials will vary with density, mean temperature, size of fibers or particles, degree and extent of bond between particles, moisture present, and the arrangement of fibers or particles within the material.

The effect of density upon conductivity (at constant mean temperature) is illustrated for two fibrous materials in Fig. 1. It will be noted that for each there is an optimum density for lowest conductivity. Typical variation of conductivity with mean temperature is shown in Fig. 2.

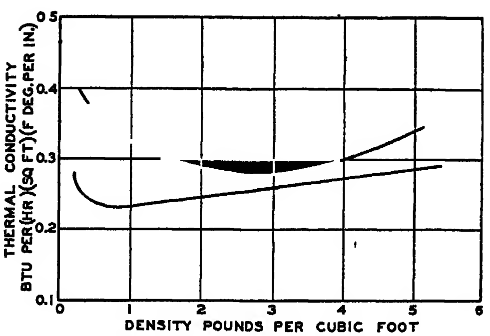


FIG. 1. TYPICAL VARIATION OF THERMAL CONDUCTIVITY WITH DENSITY—FOR FIBROUS MATERIAL

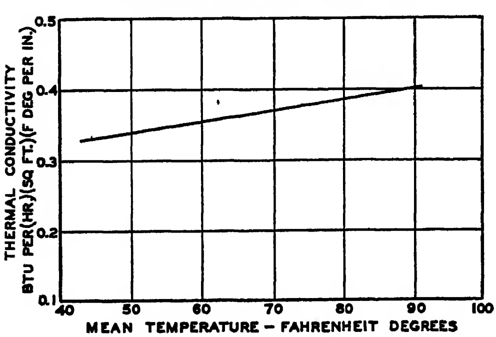


FIG. 2. TYPICAL VARIATION OF THERMAL CONDUCTIVITY WITH MEAN TEMPERATURE

TABLE 1. CONDUCTANCES (*C*) FOR SURFACES AND AIR SPACES

All conductance values expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference).

Section A. Surface Conductances for Still Air[a]

POSITION OF SURFACE	DIRECTION OF HEAT FLOW	SURFACE EMISSIVITY	
		ε = 0.83	ε = 0 05
Horizontal	Upward	1.95	1 16
Horizontal	Downward	1 21	0.44
Vertical		1.52*	0.74

Section B. Conductance of Vertical Spaces at Various Mean Temperatures[b]

MEAN TEMP FAHR DEG	CONDUCTANCES OF AIR SPACES FOR VARIOUS WIDTHS IN INCHES						
	0 128	0.250	0.364	0 493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2 385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2 650	1.590	1.390	1.295	1 210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2 819	1.702	1.492	1.390	1.295	1 280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3 078	1.870	1.650	1.534	1.425	1 402	1.392
120	3 167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1 980	1.750	1 630	1.510	1.485	1.475
140	3 340	2 035	1.800	1.680	1.550	1.530	1.519
150	3 425	2 090	1.852	1.728	1.592	1.569	1.559

Section C. Conductances and Resistances of Air Spaces Faced on One Surface with Reflective Insulation[c]

LOCATION AND POSITION OF AIR SPACE	DIRECTION OF HEAT FLOW	TEMP[d] DIFF FAHR DEG		CONDUCTANCE[e] (*C*)			RESISTANCE[e] $\left(\frac{1}{C}\right)$		
		Winter	Summer	No. of Air Spaces			No. of Air Spaces		
				1	2	3	1	2	3
Rafter Space (8 in.)									
Horizontal	Down	45			0.10	0 07		10.00	14.29
Horizontal	Up	45			0.27	0.17		3.70	5.88
Horizontal	Down		25		0.09	0 06		11.11	16 67
Horizontal	Up		25		0.24	0.16		4.17	6.25
30 deg slope	Down	45			0.15	0.10		6.67	10 00
30 deg slope	Up	45			0.25	0.17		4 00	5.88
30 deg slope	Down		25		0.13	0.09		7 69	11.11
30 deg slope	Up		25		0.23	0.14		4 35	7.14
Stud Space (3⅝ in)									
Vertical[f]		30		0.34			2.94		
Vertical		40			0 23	0.13		4 35	7.69
Vertical[f]			15	0.32			3.13		
Vertical			20		0 18	0.11		5 56	9 09
Vertical[g]		30		0 46			2 17		

[a]Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson (A S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p 513).

[b]A S H V.E. Research Report No 825—Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V E. TRANSACTIONS, Vol. 35, 1929, p. 165).

[c]Thermal Test Coefficients of Aluminum Insulation for Buildings, by G B Wilkes, F. G. Hechler and E. R. Queer (A.S H.V.E. TRANSACTIONS, Vol. 46, 1940).

[d]Temperature difference is based on total space between plaster base and sheathing, flooring or roofing.

[e]These air space conductance and resistance values are based on one reflective surface (aluminum) having an emissivity of 0 05 facing each space and are based on total space between plaster base and sheathing, flooring or roofing. The rafter and stud spaces are divided into equal spaces.

[f]Stud space is lined on plaster base side with loose paper with aluminum on surface facing air space. The resistance of the small air space between the plaster base and paper was 0 43.

[g]Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p 351).

*The recommended surface conductance for calculating heat losses for still air for non-reflective surfaces is 1.65 Btu. For a 15 mph wind velocity, the recommended value is 6.0 Btu. These coefficients were derived from Fig. 3 which was based on tests conducted at the University of Minnesota, and apply to vertical surfaces.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
BUILDING BOARDS (NON-INSULATING)	Compressed cement and asbestos sheets	123	86	2 70	....	0 37	......	(1)
	Corrugated asbestos board	20 4	110	0 48	....	2 08	......	(2)
	Pressed asbestos mill board	60.5	86	0.84	..	1.19	......	(3)
	Gypsum board—gypsum between layers of heavy paper	62.8	70	1.41	....	0.71	......	(3)
	⅜ in. gypsum board	... ..		... ...	3 73	......	0.27	
	½ in. gypsum board	.. ..	....	.....	2.82	......	0.35	
	½ in. gypsum board	53.5	90	......	2 60	......	0.38	(1)
FRAME CONSTRUCTION COMBINATIONS	1 in fir sheathing and building paper	..	30	.. ..	0.86	......	1 16	(4)
	1 in fir sheathing, building paper and yellow pine lap siding	.. ..	20	.... ..	0.50	......	2 00	(4)
	1 in. fir sheathing, building paper and stucco	.. ..	20	......	0.82	......	1.22	(4)
	Pine lap siding and building paper, siding 4 in. wide	......	16	......	0.85	......	1 18	(4)
	Yellow pine lap siding	.. ...	....	......	1.28	... ..	0 78	(4)
MASONRY MATERIALS BRICK	Damp or wet	......	....	5.0[e]	....	0.20	......	(2)
	Common yellow clay brick[a]	......	....	4.8	....	0.21	......	(4)
	One tier yellow common clay brick, one tier face brick, approx. 8 in. thick[a]	......	....	.... ..	0 77	......	1.30	(4)
CLAY TILE, HOLLOW	2 in. Tile, ½ in. plaster both sides	120 0	110	....	1.00	......	1 00	(2)
	4 in. Tile, ½ in plaster both sides	127.0	100	....	0 60	......	1 67	(2)
	6 in. Tile, ½ in. plaster both sides	124.3	105	...	0.47	.. ..	2.13	(2)
	8 in. Tile, average of 8 types (Walls No. 59, 63, 64, 66, 67, 90, 91, 92[a])	......	....		0.52	......	1.92	(4)
	12 in. Clay tile wall 8 in. x 5 in. x 12 in. and 4 in. x 5 in x 12 in.[a]	......	...	. ...	0.26	......	3.84	(4)
CONCRETE	Sand and gravel aggregate, various ages and mixes.	....	..	11.35 to 16.36	....	0.09 to 0 06	......	(5)
	Sand and gravel aggregate	142	75	12 6	..	0 08	......	(4)
	Limestone aggregate	132	75	10.8		0 09	......	(4)
	Cinder aggregate	97	75	4.9	...	0.22	......	(4)
	Steam treated limestone slag aggregate[a]	74.6	75	2.27	..	0 44	......	(4)
	Pumice (Mined in California) aggregate[a]	65 0	75	2 42	..	0 41	......	(4)
	Expanded burned clay aggregate[a]	59 9	75	2.28	....	0 44	......	(4)
	Burned clay aggregate[a]	67.1	75	2.86	....	0.35	......	(4)
	Blast furnace slag aggregate	76.0	70	1.6	..	0 63	......	(3)
	Expanded vermiculite aggregate	20	90	0 68	....	1.47	......	(3)
	Expanded vermiculite aggregate	26.7	90	0.76	....	1.32	......	(3)

AUTHORITIES:

[1]U. S Bureau of Standards, tests based on samples submitted by manufacturers.

[2]A. C. Willard, L. C. Lichty and L. A. Harding, tests conducted at the University of Illinois.

[3]J C Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers

[4]F. B. Rowley, et al, tests conducted at the University of Minnesota.

[5]A S H V E. Research Laboratory.

[6]E. A. Allcut, tests conducted at the University of Toronto.

[a]See Thermal Conductivity of Building Materials, by F. B. Rowley and A. B. Algren (University of Minnesota *Engineering Experiment Station Bulletin* No. 12).

[b]Heat Transmission Through Insulation as Affected by Orientation of Wall, by F. B Rowley and C. E Lund (A S H.V E TRANSACTIONS, Vol. 49, 1943, p 331).

[c]The Effect of Convection in Ceiling Insulation, by G. B. Wilkes and L. R. Vianey (A.S H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 196).

[d]See A.S.H V E RESEARCH REPORT No. 915—Conductivity of Concrete, by F C. Houghten and Carl Gutberlet (A.S H V E. TRANSACTIONS, Vol. 38, 1932, p. 47)

[e]See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.

[f]See BMS13, U. S. Department of Commerce, National Bureau of Standards, Washington, D. C

[g]Roofing, 0.15 in thick (1.34 lb per square foot), covered with gravel (0 83 lb per square foot), combined thickness assumed 0.25.

Table 2. Conductivities (k) and Conductances (C) of Building and Insulating Materials—Continued

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference) Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

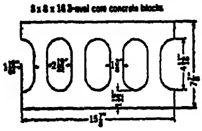
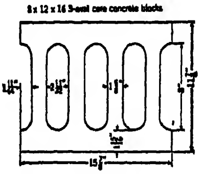
Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS —(Continued) Concrete—(Continued)	Expanded vermiculite aggregate	35	90	0 86	...	1.16	...	(3)
	Expanded vermiculite aggregate	50	90	1.10	...	0.91	...	(3)
	Concrete plank	76	75	2.5	...	0.40	...	(3)
	Cellular concrete	40 0	75	1.06	...	0 94	...	(3)
	Cellular concrete	50 0	75	1.44	...	0.69	...	(3)
	Cellular concrete	60 0	75	1.80	...	0 56	...	(3)
	Cellular concrete	70 0	75	2.18	...	0 46	...	(3)
8 In. Concrete Blocks 8 x 8 x 16 3-oval core concrete blocks 	8 in. three oval core, sand and gravel aggregate[a]	126.4	40	...	0.90	...	1.11	(4)
	8 in. three oval core, crushed limestone aggregate[a]	134.3	40	...	0.86	...	1.16	(4)
	8 in. three oval core, cinder aggregate[a]	86.2	40	...	0.58	...	1 73	(4)
	8 in. three oval core, burned clay aggregate[a]	67.7	40	...	0.50	...	2.00	(4)
	8 in. three oval core, expanded blast furnace slag aggregate[a]	...	40	...	0.49	...	2 04	(4)
12 In. Concrete Blocks 8 x 12 x 16 3-oval core concrete blocks 	12 in. three oval core, sand and gravel aggregate[a]	124 9	40	...	0 78	...	1.28	(4)
	12 in three oval core, cinder aggregate[a].	86.2	40	...	0.53	...	1.88	(4)
	12 in. three oval core, burned clay aggregate[a]	76.7	40	...	0.47	...	2 13	(4)
Gypsum	3 in. solid gypsum partition tile[a]	...	...	2.41	...	0 42	...	(4)
	3 in. three cell gypsum partition tile[a]	...		...	0 74	...	1.35	(4)
	4 in. three cell gypsum partition tile[a]	...	...	...	0.60	...	1.67	(4)
	87½ per cent gypsum, 12½ per cent wood chips	51.2	74	1.66	...	0 60	...	(4)
	Gypsum plaster	...	...	3 30	...	0.30	...	
PLASTERING MATERIALS	Gypsum plaster, ⅜ in. thick	...	73	...	8.80	...	0 11	(4)
	Cement plaster	...	...	8.00	...	0 13	...	(2)
	Wood, lath and plaster, total thickness ¾ in.	...	70	...	2.50	...	0 40	(4)
	Gypsum plaster and expanded vermiculite, 4 to 1 mix	39.9	75	0.85	...	1.18	...	(3)
	Insulating plaster 0.9 in. thick applied to ⅜ in. gypsum board	54 0	75	...	1.07	...	0.93	(3)
ROOFING	Asbestos shingles	65 0	75	...	6.0	...	0.17	(3)
	Asphalt, composition or prepared.	70 0	75	...	6.5	...	0 15	(3)
	Asphalt shingles	70 0	75	...	6.5	...	0.15	(3)
	Built-up roofing, bitumen or felt, gravel or slag surfaced[b]	...	...	1.33	...	0 75	...	(2)
	Slate	...	...	10.00		0.10	...	
	Wood shingles	...	...	...	1 28	...	0 78	
WOODS	Balsa	20 0	90	0.58	...	1.72	...	(1)
	Balsa	8.8	90	0.38	...	2.63	...	(1)
	Balsa	7.3	90	0.33	...	3.03	...	(1)
	California redwood, 0 per cent moisture[a]	28.0	75	0 70	...	1 43	...	(4)
	Cypress	28.7	86	0 67	...	1.49	...	(1)
	Douglas fir, 0 per cent moisture[a]	34.0	75	0 67	...	1 49	...	(4)
	Eastern hemlock, 0 per cent moisture[a]	30.0	75	0.76	...	1.32	...	(4)
	Long leaf yellow pine, 0 per cent moisture[a]	40.0	75	0.86	...	1.16	...	(4)
	Mahogany	34.3	86	0.90	...	1.11	...	(1)

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS—Continued

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
)DS—(Continued)	Hard maple, 0 per cent moisture[a]	46.0	75	1.05	.	0.95	——	(4)
	Maple	44.3	86	1.10		0.91	——	(1)
	Maple, across grain	40.0	75	1.20	..	0.83	——	(3)
	Norway pine, 0 per cent moisture[a]	32.0	75	0.74	.	1.35	——	(4)
	Red cypress, 0 per cent moisture[a]	32.0	75	0.79		1.27	——	(4)
	Red oak, 0 per cent moisture[a]	48.0	75	1.18		0.85	——	(4)
	Short leaf yellow pine, 0 per cent moisture[a]	36.0	75	0.91	...	1.10	——	(4)
	Soft elm, 0 per cent moisture[a]	34.0	75	0.88		1.14	——	(4)
	Soft maple, 0 per cent moisture[a]	42.0	75	0.95	..	1.05	——	(4)
	Sugar pine, 0 per cent moisture[a]	28.0	75	0.64		1.56	——	(4)
	Virginia pine	34.3	86	0.96	....	1.04	——	(1)
	West coast hemlock, 0 per cent moisture[a]	30.0	75	0.79		1.27	——	(4)
	White pine	31.2	86	0.78	...	1.28	——	(1)
	Yellow pine			1.00	.	1.00	——	(3)
	Sawdust, various	12.0	90	0.41		2.44	——	(1)
	Shavings, various from planer	8.8	90	0.41		2.44	——	(1)
	Shavings, from maple beech and birch (coarse)	13.2	90	0.36	...	2.78	——	(1)
ULATING MATERIALS ANKET AND BAT INSULATIONS	Chemically treated wood fibers held between layers of strong paper	3.62	70	0.25	....	4.00	——	(3)
	Eel grass between strong paper	4.60	90	0.26	...	3.85	——	(1)
	Eel grass between strong paper	3.40	90	0.25		4.00	——	(1)
	Flax fibers between strong paper	4.90	90	0.28	..	3.57	——	(1)
	Chemically treated hog hair between kraft paper	5.76	71	0.26	..	3.85	——	(3)
	Chemically treated hog hair between kraft paper and asbestos paper	7.70	71	0.28		3.57	——	(3)
	Hair felt between layers of paper	11.00	75	0.25	..	4.00	——	(3)
	Kapok between burlap or paper	1.00	90	0.24	..	4.17	——	(1)
	Stitched and creped expanding fibrous blanket	1.50	70	0.27		3.70	——	(3)
	Paper and asbestos fiber with emulsified asphalt binder	4.2	94	0.28		3.57	——	(1)
	Cotton insulating bat	0.875	72	0.24		4.17	——	(3)
	Cotton fibers							
	Short Staple Linters, Fireproofed	6.25	90	0.25	..	4.00	——	(1)
	Short Staple Linters, Fireproofed	4.50	90	0.24	..	4.17	——	(1)
	Short Staple Linters, Fireproofed	2.45	90	0.24	..	4.17	——	(1)
	Short Staple Linters, Fireproofed	1.60	90	0.26	..	3.85	——	(1)
	Short Staple Linters, Fireproofed	0.85	90	0.29	..	3.45	——	(1)
	Short Staple Linters, Fireproofed	0.65	90	0.30		3.33	——	(1)
	Felted cattle hair	13.00	90	0.26	..	3.84	——	(1)
	Felted cattle hair	11.00	90	0.26		3.84	——	(1)
	Felted hair and asbestos	7.80	90	0.28	..	3.57	——	(1)
	Ground paper between two layers, each ⅜ in thick made up of two layers of kraft paper (sample ¾ in thick)	12.1	75		0.40	——	2.50	(4)
EFLECTIVE	See Table 1, Section C	.....	..	.....	....	——	——	
SULATING BOARD	Made from sugar cane fiber	13.5	70	0.33		3.03	——	(3)
	Made from corn stalks	15.00	71	0.33	...	3.03	——	(3)
	Made from exploded wood fibers	17.90	73	0.32	..	3.12	——	(4)
	Made from hard wood fibers	15.20	70	0.32	..	3.12	——	(3)
	Made from wood fiber	15.90	72	0.33	..	3.03	——	(3)
	Made from wood fiber	15.00	70	0.33	..	3.03	——	(3)
	Made from wood fiber		52	0.33	..	3.03	——	(6)
	Made from wood fiber	8.50	72	0.29	....	3.45	——	(3)
	Made from wood fiber	15.20	..	0.33	...	3.03	——	(3)
	Made from wood fiber	16.90	90	0.34	..	2.94	——	(1)

See notes on Page 117.

Table 2. Conductivities (k) and Conductances (C) of Building and Insulating Materials—Concluded

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference) Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
INSULATING MATERIALS—Continued								
Insulating Board—Continued	Made from licorice root	16.1	81	0.34		2 94	......	(3)
	½ in. insulating boards without special finish[f] (eleven samples)	16.5 to 21.8	90	0.33 to 0.40	.. ..	3 03 to 2.50	......	(1)
	1 in. insulating board[e]	13.2	...	0.34	....	2.94	......	(4)
Loose Fill Type	Made from ceiba fibers	1.90	75	0.23	..	4.35	......	(3)
	Made from ceiba fibers	1.60	75	0.24	..	4 17	......	(3)
	Fibrous material made from dolomite and silica	1.50	75	0.27	..	3 70	......	(3)
	Fibrous material made from slag	9 40	103	0.27		3 70	......	(1)
	Redwood bark	3 00	90	0.31		3.22	......	(1)
	Redwood bark	5.00	75	0.26	..	3.84	......	(3)
	Glass wool fibers 0.0003 in. to 0.006 in. in diameter	1.50	75	0.27		3 70	......	(3)
	Granular insulation made from combined silicate of lime and alumina	4.20	72	0.24		4 17	......	(3)
	Expanded vermiculite	....	...	0.48	...	2 08	......	(1)
	Expanded vermiculite, particle size—$-3+14$	6.2		0.32	...	3 12	......	(3)
	Regranulated cork about $\frac{3}{16}$ in. particles	8 10	90	0.31		3.22	......	(1)
	Hand applied granular mineral wool 2 in. to 6 in. thick; horizontal position[b]. No covering	6 05 to 7.13	....	0.30 to 0.33	.. ....	3.33 to 3.03	......	(4)
	4 in. machine blown granular mineral wool, horizontal position[b]. No covering	5.74	..	0.30	...	3.33	......	(4)
	Rock wool	10.0	90	0.27	..	3 70	......	(1)
Slab Insulations	Corkboard, no added binder	14 0	90	0.34	...	2 94	......	(1)
	Corkboard, no added binder	10 6	90	0.30	..	3.33	......	(1)
	Corkboard, no added binder	7 0	90	0.27	....	3 70	......	(1)
	Corkboard, no added binder	5 4	90	0.25		4.00	......	(1)
	Corkboard[e]	8.7		0.29		3 45	......	(4)
	Corkboard, asphaltic binder	14.5	90	0.32	..	3.12	......	(1)
	Chemically treated hog hair with film of asphalt	10.0	75	0.28	....	3.57	......	(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.8	70	0.30	....	3.33	......	(3)
	Made from 85 per cent magnesia and 15 per cent asbestos	19.3	86	0.51	....	1 96	......	(1)
	Made from shredded wood and cement	24.2	72	0 46	.. .	2 17	......	(3)
	Made from shredded wood and cement[e]	29.8	....	0 77	....	1.30	......	(4)

See notes on Page 117.

Surface Conductance

The surface conductance of a wall is the combined heat transfer to or from the wall by radiation, convection and conduction. Each of the three portions making up the total may vary independently of the others, thus affecting the total conductance. The heat transfer by radiation between two surfaces is controlled by the character of the surfaces (emissivity), the temperature difference between them, and the solid angle through which they *see* each other. The heat transfer by convection and conduction is controlled by the roughness of the surface, by air movement and temperature difference between the air and the surface.

The importance of the effect of temperature of *surrounding surfaces* on the surface conductance due to the effect on radiation is illustrated in

Table 3, which applies to a vertical surface at 80 F, with ambient air at 70 F and effective emissivity equal to 0.83 [3].

In many cases, because the heat resistance of the internal parts of the wall is high compared with the surface resistance, the surface factors are of minor importance. In other cases, *e.g.* single glass windows, the surface resistances constitute almost the entire resistance and are therefore very important. In a building heated by convection there is only a slight difference between the temperatures of the interior wall surface and the surroundings, but if the building is heated by radiant panels there may be a considerable difference [4]. (See also Chapter 31.)

The convection part of the surface conductance coefficient is affected markedly by air movement. This is illustrated by Fig. 3, which shows the surface conductances for different materials at a mean temperature of 20 F and for wind velocities up to 40 mph. These include the radiation portion of the coefficient, which for ordinary building materials under these conditions would be constant at about 0.7 Btu.

TABLE 3. VARIATION IN SURFACE CONDUCTANCE COEFFICIENT WITH DIFFERENT TEMPERATURES OF SURROUNDING SURFACE

SURROUNDING SURFACE TEMPERATURE	75 F	70 F	69 F	60 F	50 F
Convection—Btu per (hr) (sq ft)	6.6	6.6	6 6	6.6	6.6
Radiation—Btu per (hr) (sq ft)	4.4	8.6	9.6	17.0	24.9
Total—Btu per (hr) (sq ft)	11.0	15.2	16.2	23.6	31.5

Due to these variations for different conditions the selection of surface conductance coefficients for a practical building becomes a matter of judgment. In calculating the over-all heat transmission coefficients for the walls, etc. of Tables 5 to 18, 1.65 has been selected as an *average inside surface conductance* and 6.0 as an *average outside surface conductance* for a 15-mile wind. These values apply only to ordinary building materials and should not be used for bright metal surfaces having a low emissivity.

In special cases, where surface conductance coefficients become important factors in the over-all rates of heat transfer, more selective coefficients may be required. The surface conductance values given in Table 1, **Section A** are based on recent tests and are for still air conditions and surface emissivities of 0.83 and 0.05 respectively, and may be used where it is desirable to differentiate between horizontal and vertical surfaces or where coefficients applicable to low-emissivity surfaces are required.

Air Space Conductance

The transfer of heat across an air space involves the boundary surfaces as well as the intervening air, consequently the factors influencing surface conductance play an important part in determining the conductance of the air space. The coefficients given for air space conductance represent the total conductance from surface to surface.

The radiation portion of the coefficient is affected by the difference in temperature between the boundary surfaces and by their respective

emissivities and is practically independent of depth. The convection and conduction transfer is controlled by depth and shape of the air space, the roughness of the boundary surfaces, the mean temperature and the direction of heat flow. For air spaces usually employed in building construction, the radiation and convection factors vary independently of each other.

Table 1, **Section B** gives experimentally-determined conductances of vertical air spaces bounded by such materials as paper, wood, plaster, etc., having emissivity coefficients of 0.8 or higher, and having extended parallel surfaces perpendicular to the direction of heat flow. The conductances decrease as the depth is increased but change only slightly

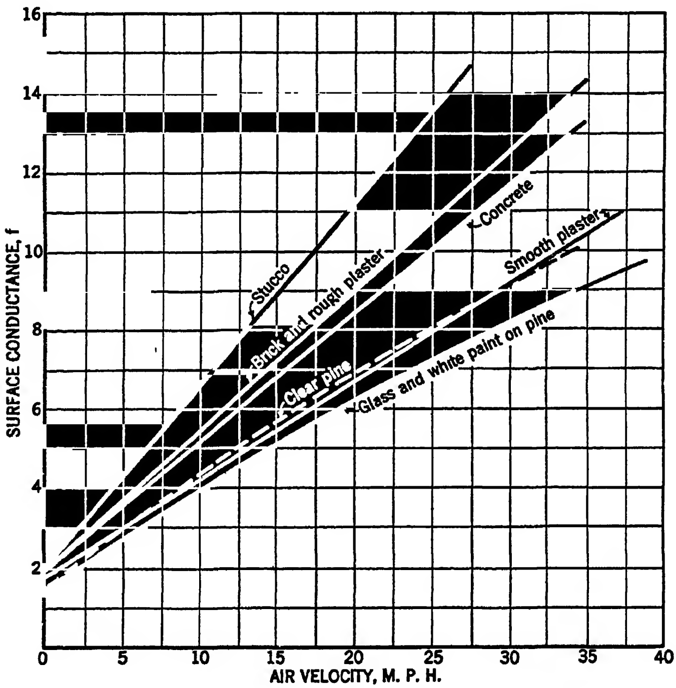


FIG. 3. CURVES SHOWING RELATION BETWEEN SURFACE CONDUCTANCES FOR DIFFERENT SURFACES AT 20 F MEAN TEMPERATURE

for spaces greater than ¾ in. Air space tests reported by Wilkes and Peterson gave conductance values for air spaces of 3⅝ in. depth having boundary surfaces with emissivity values of 0.83 as follows [5].

Vertical..1.17
Horizontal (heat flow upward)..............1 32
Horizontal (heat flow downward).......0.94

Since, in buildings, the same constructions may be used for conditions where the direction of heat flow may be in one direction or its opposite, and since much of the construction involves vertical air spaces, an average value of 1.10 Btu per (hour) (square foot) (Fahrenheit degree temperature difference) was chosen for use in calculating the over-all coefficients in Tables 5 to 18 wherever air spaces ¾ in. or more in depth were involved.

If one or both boundary surfaces of an air space are faced with metals which have low emissivity surfaces, the radiant heat transfer will be greatly reduced in comparison with that occurring from surfaces of ordinary building materials. Table 1, **Section C** gives conductances and resistances of air spaces *bounded by one reflective surface* with an emissivity of 0.05. These values include heat transferred both by radiation and convection, but the radiation component is relatively small for the test conditions.

When reflective materials are installed with single or multiple air spaces, the position (vertical, horizontal or inclined) of the material and the direction of heat flow must be taken into consideration. For example, the resistance to *upward* heat flow is about one-third the resistance to *downward* heat flow in a horizontal position (Table 1, **Section C**). The difference between the conductance through vertical air spaces and that through horizontal and sloping air spaces with upward heat flow is considerably less. For upward heat flow it is recommended that a value of 0.46 be used for the conductance of horizontal or sloping air spaces bounded on one side by reflective materials having an emissivity of approximately 0.05. The same conductance value is also recommended for similar vertical air spaces.

When considering heat transfer to and from reflective surfaces in building construction, the emissivity should be known. This can be determined directly for the long wave length radiation corresponding to average room and wall temperatures. The possibility of change in emissivity with time of exposure due to surface coatings, chemical action, deposition of dust, etc. must be considered in selecting a material for use [6].

PRACTICAL COEFFICIENTS AND THEIR USE

For practical purposes it is necessary to have average coefficients that may be applied to various materials and types of construction without the necessity of making actual tests. In Table 2 coefficients are given for a group of materials which have been selected from tests by various authorities. Since there is some variation in the resulting values due to variations in materials and in test conditions, average values for the usual conditions encountered in building practice have been selected and listed in Table 4. These coefficients were used in the calculation of over-all coefficients given in Tables 5 to 18. These tables constitute typical examples of combinations frequently used, but any special constructions not given can be computed by the use of the conductivity values in Table 4 and the fundamental heat transfer formulae.

Caution

The user should realize that the average conductivity and conductance values given in Tables 2 or 4 do not necessarily apply to all products of the same general description. In using these values judgment should be exercised with regard to the extent to which the product (either as received or as applied) will comply with the tabulated values. Exact conductivities or conductances for specific materials should be obtained from the manufacturer.

Insulating Materials

In order to determine the benefit derived from the addition of insulating materials to a given construction, the over-all coefficient of heat

TABLE 4 CONDUCTIVITIES (k) AND CONDUCTANCES (C) USED IN CALCULATING HEAT TRANSMISSION COEFFICIENTS (U) IN TABLES 5 TO 18

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

MATERIAL	DESCRIPTION	CONDUCTIVITY OR CONDUCTANCE		RESISTANCE	
		(k)	(C)	Per Inch Thickness $\left(\frac{1}{k}\right)$	For Thickness Listed $\left(\frac{1}{C}\right)$
AIR SPACES					
Bounded by ordinary materials	Vertical[a], ¾ in. or more in width		1.10		0.91
Bounded by aluminum foil	Vertical[a], ¾ in. or more in width		0.46		2.17
EXTERIOR FINISHES (Frame Walls)					
Brick Veneer	4 in. thick (nominal)		2.27		0.44
Stucco (1 in.)		12.50		0.08	
Wood Shingles			1.28		0.78
Yellow Pine Lap Siding			1.28		0.78
INSULATING MATERIALS					
Aluminum Foil	See Air Spaces				
Bats and Blankets	Made from mineral or vegetable fiber or animal hair, enclosed or open	0.27		3.70	
Corkboard	Pure, no added binder	0.30		3.33	
Insulating Board	Vegetable fiber	0.33		3.03	
Mineral Wool	Fiber made from rock, slag or glass	0.27		3.70	
Vermiculite	Expanded	0.48		2.08	
INTERIOR FINISHES					
Composition Wallboard	3/16 in. to 3/8 in. thick	0.50		2.00	
Gypsum Plaster		3.30		0.30	
Gypsum Board (⅜ in.)	Plain or decorated		3.70		0.27
Gypsum Lath (⅜ in.) and Plaster	Plaster thickness assumed ½ in.		2.4		0.42
Insulating Board (½ in.)	Plain or decorated		0.66		1.52
Insulating Board Lath (½ in.) and Plaster	Plaster thickness assumed ½ in.		0.60		1.67
Insulating Board Lath (1 in.) and Plaster	Plaster thickness assumed ½ in.		0.31		3.18
Metal Lath and Plaster	Plaster thickness assumed ¾ in.		4.40		0.23
Plywood (⅜ in.)	Plain or decorated		2.12		0.47
Wood Lath and Plaster			2.50		0.40
MASONRY MATERIALS					
Brick	Adobe, assumed 4 in. thick		0.89		1.12
Brick	Common, assumed 4 in. thick		1.25		0.80
Brick	Face, assumed 4 in. thick		2.30		0.43
Cement Mortar		12.00		0.08	
3 in. Clay tile (hollow)			1.28		0.78
4 in. Clay tile (hollow)			1.00		1.00
6 in. Clay tile (hollow)			0.64		1.57
8 in. Clay tile (hollow)			0.60		1.67
10 in. Clay tile (hollow)			0.58		1.72
12 in. Clay tile (hollow)			0.40		2.50
16 in. Clay tile (hollow)			0.31		3.23
Concrete	Light weight aggregate[b]	2.50		0.40	
Concrete	Sand and gravel aggregate	12.00		0.08	
3 in. Concrete blocks	Hollow, cinder aggregate		1.28		0.78
4 in. Concrete blocks	Hollow, cinder aggregate		1.00		1.00
8 in. Concrete blocks	Hollow, gravel aggregate		1.00		1.00
12 in. Concrete blocks	Hollow, gravel aggregate		0.80		1.25
8 in. Concrete blocks	Hollow, cinder aggregate		0.60		1.66
12 in. Concrete blocks	Hollow, cinder aggregate		0.53		1.88
8 in. Concrete blocks	Hollow, light weight aggregate[b]		0.50		2.00
12 in. Concrete blocks	Hollow, light weight aggregate[b]		0.47		2.13
Gypsum fiber concrete	87½ per cent gypsum and 12½ per cent wood chips	1.66		0.60	
3 in. Gypsum tile	Hollow		0.61		1.64
4 in. Gypsum tile	Hollow		0.46		2.18
Stucco		12.50		0.08	
Tile and Terrazzo	For flooring	12.00		0.08	
Stone		12.50		0.08	

[a]Conductance values for horizontal air spaces depend on whether the heat flow is upward or downward, but in most cases it is sufficiently accurate to use the same values for horizontal as for vertical air spaces.

[b]Expanded slag, burned clay or pumice.

TABLE 4. CONDUCTIVITIES (k) AND CONDUCTANCES (C) USED IN CALCULATING HEAT TRANSMISSION COEFFICIENTS (U) IN TABLES 5 TO 18—Concluded

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference) Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

MATERIAL	DESCRIPTION	Conductivity or Conductance (k)	(C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	For Thickness Listed $\left(\frac{1}{C}\right)$
ROOFING MATERIALS					
Asbestos Shingles			6.00		0.17
Asphalt Shingles			6.50		0.15
Built-up Roofing	Assumed thickness ⅜ in.		3.53		0.28
Heavy Roll Roofing			6.50		0.15
Slate	Assumed ½ in.	10.00	20.00	0.10	0.05
Wood Shingles			1.28		0.78
SHEATHING					
Gypsum (½ in.)			2.82		0.35
Insulating Board (²⁵⁄₃₂ in.)			0.42		2.37
Plywood (⁵⁄₁₆ in.)			2.56		0.39
Fir or Yellow Pine (1 in.)	Actual thickness ²⁵⁄₃₂ in.		1.02		0.98
Fir, plus building paper	Actual thickness ²⁵⁄₃₂ in.		0.86		1.16
SURFACES					
Still air	Ordinary non-reflective materials, vertical		1.65		0.61
15 mph wind velocity	Ordinary non-reflective materials, vertical		6.00		0.17
WOODS					
Fir sheathing (1 in.) building paper and Yellow Pine lap siding			0.50		2.00
Maple or Oak		1.15		0.87	
Yellow Pine or Fir		0.80		1.25	

transmission U_i of the insulated construction may be compared with the corresponding coefficient U without insulation. Attention is called to the necessity of applying the insulating material in accordance with the manufacturer's specification. The engineer must carefully evaluate the economic considerations involved in the selection of an insulating material as adapted to various building constructions. Lack of proper evaluation, or improper installation may lead to unsatisfactory results.

Computed Heat Transmission Coefficients

Computed over-all heat transmission coefficients of many common types of building construction are given in Tables 5 to 18, inclusive, each coefficient being identified by a serial number except in Table 18. For example, the coefficient U of a brick-veneer, frame wall with wood sheathing and ½ in. of plaster on gypsum lath is 0.27, (Wall No. 28-C in Table 5) and with 2 in. of blanket or bat insulation the coefficient would be 0.097 (No. 49-B in Table 6).

Example 1. Calculate the coefficient of heat transmission U of a brick-veneer, frame wall with wood sheathing, building paper, and ½ in. plaster on ⅜ in. gypsum lath, based on a wind exposure of 15 mph. Also calculate the coefficient when 2 in. of bat insulation is added, leaving an air space in the wall, and correcting for framing amounting to 15 per cent of the wall area.

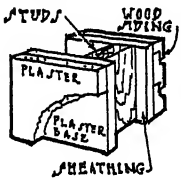
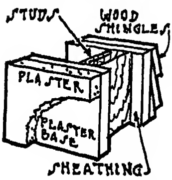
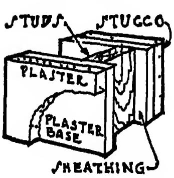
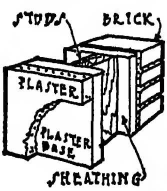
Solution: Starting from the exterior, the resistances making up the total heat resistance are (1) exterior surface, (2) brick-veneer, (3) wood sheathing ²⁵⁄₃₂ in. thick and building paper, (4) air space 3⅝ in. wide, (5) ⅜ in. gypsum lath and plaster, (6) interior surface. Then using the values from Table 4 in Equation 4:

$$U = \frac{1}{\frac{1}{6.0} + \frac{1}{2.27} + \frac{1}{0.86} + \frac{1}{1.10} + \frac{1}{2.4} + \frac{1}{1.65}}$$

Table 5. Coefficients of Transmission (*U*) of Frame Walls

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

No Insulation Between Studs[a] (See Table 6)

Exterior Finish	Interior Finish	Type of Sheathing: Gypsum (1/2 in. thick) A	Ply-wood (5/16 in. thick) B	Wood[f] (25/32 in. thick) Bldg. Paper C	Insulating Board (25/32 in thick) D	Wall Number
Wood Siding (Clapboard) 	Metal Lath and Plaster[b]	0.33	0.32	0.26	0.20	1
	Gypsum Board (3/8 in.) Decorated	0.32	0.32	0 26	0.20	2
	Wood Lath and Plaster	0.31	0.31	0.25	0 19	3
	Gypsum Lath (3/8 in.) Plastered[c]	0.31	0 30	0.25	0 19	4
	Plywood (3/8 in.) Plain or Decorated	0.30	0.30	0 24	0 19	5
	Insulating Board (1/2 in.) Plain or Decorated	0.23	0.23	0.19	0.16	6
	Insulating Board Lath (1/2 in.) Plastered[c]	0.22	0.22	0 19	0 15	7
	Insulating Board Lath (1 in.) Plastered[c]	0.17	0.17	0.15	0.12	8
Wood[d] Shingles 	Metal Lath and Plaster[b]	0.25	0.25	0.26	0 17	9
	Gypsum Board (3/8 in.) Decorated	0 25	0.25	0.26	0 17	10
	Wood Lath and Plaster	0.24	0.24	0 25	0 16	11
	Gypsum Lath (3/8 in.) Plastered[c]	0.24	0.24	0.25	0.16	12
	Plywood (3/8 in.) Plain or Decorated	0.24	0 24	0.24	0 16	13
	Insulating Board (1/2 in.) Plain or Decorated	0.19	0 19	0 19	0 14	14
	Insulating Board Lath (1/2 in.) Plastered[c]	0.19	0 18	0 19	0.13	15
	Insulating Board Lath (1 in.) Plastered[c]	0.14	0 14	0.15	0.11	16
Stucco 	Metal Lath and Plaster[b]	0.43	0 42	0.32	0.23	17
	Gypsum Board (3/8 in.) Decorated	0 42	0 41	0.31	0.23	18
	Wood Lath and Plaster	0 40	0.39	0 30	0.22	19
	Gypsum Lath (3/8 in.) Plastered[c]	0.39	0.39	0.30	0.22	20
	Plywood (3/8 in.) Plain or Decorated	0.39	0.38	0.29	0.22	21
	Insulating Board (1/2 in.) Plain or Decorated	0.27	0.27	0 22	0 18	22
	Insulating Board Lath (1/2 in.) Plastered[c]	0.26	0.26	0.22	0 17	23
	Insulating Board Lath (1 in.) Plastered[c]	0.19	0.19	0.16	0.14	24
Brick Veneer[e] 	Metal Lath and Plaster[b]	0.37	0.36	0.28	0.21	25
	Gypsum Board (3/8 in) Decorated	0.36	0.36	0.28	0.21	26
	Wood Lath and Plaster	0.35	0 34	0.27	0.20	27
	Gypsum Lath (3/8 in.) Plastered[c]	0.34	0.34	0.27	0.20	28
	Plywood (3/8 in.) Plain or Decorated	0.34	0.33	0.27	0.20	29
	Insulating Board (1/2 in.) Plain or Decorated	0.25	0.25	0.21	0 17	30
	Insulating Board Lath (1/2 in.) Plastered[c]	0.24	0.24	0.20	0 16	31
	Insulating Board Lath (1 in.) Plastered[c]	0.18	0 18	0.15	0 13	32

[a]Coefficients not weighted; effect of studding neglected.
[b]Plaster assumed 3/4 in. thick.
[c]Plaster assumed 1/2 in. thick.
[d]Furring strips (1 in. nominal thickness) between wood shingles and all sheathings except wood.
[e]Small air space and mortar between building paper and brick veneer neglected.
[f]Nominal thickness, 1 in.

TABLE 6. COEFFICIENTS OF TRANSMISSION (U) OF FRAME WALLS WITH INSULATION BETWEEN FRAMING[a, b]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

COEFFICIENT WITH *NO* INSULATION BETWEEN FRAMING	COEFFICIENT WITH INSULATION BETWEEN FRAMING				NUMBER
	MINERAL WOOL OR VEGETABLE FIBERS IN BLANKET OR BAT FORM[c] (Thickness below)			3⅝ IN. MINERAL WOOL BETWEEN FRAMING[d]	
	1 IN	2 IN.	3 IN.		
	A	B	C	D	
0.11	0.078	0.063	0.054	0.051	33
0.12	0.083	0.067	0.056	0.053	34
0.13	0.088	0.070	0.058	0.055	35
0.14	0.092	0.072	0.061	0.057	36
0.15	0.097	0.075	0.062	0.059	37
0.16	0.10	0.078	0.064	0.060	38
0.17	0.10	0.080	0.066	0.062	39
0.18	0.11	0.082	0.067	0.063	40
0.19	0.11	0.084	0.069	0.065	41
0.20	0.12	0.086	0.070	0.066	42
0.21	0.12	0.088	0.072	0.067	43
0.22	0.12	0.089	0.073	0.068	44
0.23	0.12	0.091	0.074	0.069	45
0.24	0.12	0.093	0.075	0.070	46
0.25	0.13	0.094	0.076	0.071	47
0.26	0.13	0.096	0.077	0.072	48
0.27	0.14	0.097	0.078	0.073	49
0.28	0.14	0.098	0.079	0.073	50
0.29	0.14	0.10	0.080	0.075	51
0.30	0.14	0.10	0.080	0.075	52
0.31	0.14	0.10	0.081	0.076	53
0.32	0.15	0.10	0.082	0.077	54
0.33	0.15	0.10	0.083	0.077	55
0.34	0.15	0.10	0.083	0.078	56
0.35	0.15	0.11	0.084	0.078	57
0.36	0.15	0.11	0.085	0.079	58
0.37	0.16	0.11	0.085	0.080	59
0.38	0.16	0.11	0.086	0.080	60
0.39	0.16	0.11	0.086	0.081	61
0.40	0.16	0.11	0.087	0.082	62
0.41	0.16	0.11	0.087	0.082	63
0.42	0.16	0.11	0.088	0.082	64
0.43	0.17	0.11	0.088	0.082	65
0.44	0.17	0.11	0.089	0.083	66

[a]This table may be used for determining the coefficients of transmission of frame constructions with the types and thicknesses of insulation indicated in Columns A to D inclusive between framing. Columns A, B and C may be used for walls, ceilings or roofs with only one air space between framing but are not applicable to ceilings with *no* flooring above. (See Table 11.) Column D is applicable to walls only. *Example:* Find the coefficient of transmission of a frame wall consisting of wood siding, 25/32 in insulating board sheathing, studs, gypsum lath and plaster, with 2 in. blanket insulation between studs. According to Table 5, a wall of this construction with *no insulation* between studs has a coefficient of 0.19 (Wall No. 4D). Referring to Column B above, it will be found that a wall of this value with 2 in. blanket insulation between the studs has a coefficient of 0.084.

[b]Coefficients corrected for 2 x 4 framing, 16 in. on centers—15 per cent of surface area.

[c]Based on one air space between framing.

[d]No air space.

$$= \frac{1}{0.17 + 0.44 + 1.16 + 0.91 + 0.42 + 0.61} = \frac{1}{3.71} = 0.27$$

When 2 in. of bat insulation is added in the air space in the wall, an air space will still be left since it was originally $3\frac{5}{8}$ in. wide. Then,

$$U = \frac{1}{3.71 + \frac{2}{0.27}} = \frac{1}{11.11} = 0.090$$

This coefficient (0.090) applies only to the wall where no studding or framing is present and is assumed to constitute 85 per cent of the whole wall. Then for 1 sq ft of wall the heat transfer through the insulated portion is 0.85×0.090 or 0.0765 Btu.

In the portion of wall where framing is present, the air space will be replaced with an equivalent thickness of wood framing. Then,

$$U \text{ (through framing)} = \frac{1}{\frac{1}{6.0} + \frac{1}{2.27} + \frac{1}{0.86} + \frac{3\,625}{0.80} + \frac{1}{2.4} + \frac{1}{1.65}}$$

$$U = 0.136$$

Then for 1 sq ft of wall the heat transfer through the framing is 15 per cent of 0.136 or 0.0204 Btu. The total heat transfer through 1 sq ft of wall (with $3\frac{5}{8}$ in. framing covering 15 per cent of area) equals

$$U \text{ (corrected)} = 0.0765 + 0.0204 = 0.097 \text{ Btu}$$

This is the value shown for No. 49-B in Table 6, which is included to eliminate the need for the calculation of framing corrections when insulation is used in frame construction.

In making the calculations for values of U shown in Tables 5 to 18, the following conditions have been assumed:

Equilibrium or steady-state heat transfer, eliminating effects of heat capacity.

Surrounding surfaces at ambient air temperatures.

Exterior wind velocity of 15 mph.

Surface emissivity of ordinary building materials = 0 83.

No correction for position or direction of heat flow. (Average coefficients used).

Air spaces are $\frac{3}{4}$ in. or more in width.

Variation of conductivity with mean temperature neglected.

Corrections for framing made on basis of parallel heat flow through 2×4 in. (nominal) studs, 16 in. on centers, the framing covering 15 per cent of wall area.

Actual thicknesses of lumber assumed to be as follows:

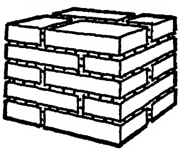
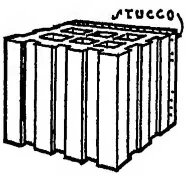
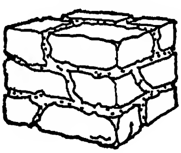
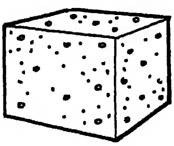
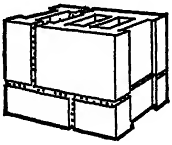
Nominal	*Actual*
1 in. (S-2-S)	$\frac{25}{32}$ in.
$1\frac{1}{2}$ in. (S-2-S)	$1\frac{5}{16}$ in.
2 in. (S-2-S)	$1\frac{5}{8}$ in.
$2\frac{1}{2}$ in. (S-2-S)	$2\frac{1}{8}$ in.
3 in. (S-2-S)	$2\frac{5}{8}$ in.
4 in. (S-2-S)	$3\frac{5}{8}$ in.
Finish flooring, (maple or oak)	$\frac{13}{16}$ in.

Coefficients for frame construction are corrected for the effect of framing where such correction would increase the coefficients, but not where the correction would decrease the coefficients [7].

It should be noted that the effects of poor workmanship in construction and installation have an increasingly greater percentage effect on heat transmission as the coefficient becomes numerically smaller. Failure to meet design estimates may be caused by lack of proper attention to

TABLE 7. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

Type of Masonry		Thickness of Masonry Inches	Interior Finish (Plus Insulation Where Indicated): Plain Walls—No Interior Finish	Plaster (½ in.) on Walls	Metal Lath and Plaster[f] Furred[h]	Gypsum Board (⅜ in.) Decorated—Furred[h]	Gypsum Lath (⅜ in.) Plastered[g]—Furred[h]	Insulating Board (½ in.) Plain or Decorated—Furred[h]	Insulating Board Lath (½ in.) Plastered[g]—Furred[h]	Insulating Board Lath (1 in.) Plastered[g]—Furred[h]	Gypsum Lath[g] Plastered Plus 1 in. Blanket Insulation—Furred[h]	Wall Number
			A	B	C	D	E	F	G	H	I	
Solid[a] Brick		8	0.50	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	67
		12	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.14	0.13	68
		16	0.28	0.27	0.21	0.21	0.20	0.17	0.16	0.13	0.12	69
Hollow[b] Tile (Stucco Exterior Finish)		8	0.40	0.37	0.27	0.27	0.26	0.20	0.20	0.15	0.13	70
		10	0.39	0.37	0.27	0.27	0.26	0.20	0.19	0.15	0.13	71
		12	0.30	0.28	0.22	0.22	0.21	0.17	0.17	0.13	0.12	72
		16	0.24	0.24	0.19	0.19	0.18	0.15	0.15	0.12	0.11	73
Stone[c]		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	74
		12	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	75
		16	0.49	0.45	0.31	0.31	0.29	0.22	0.22	0.16	0.14	76
		24	0.37	0.35	0.26	0.26	0.25	0.19	0.19	0.15	0.13	77
Poured Concrete[d]		6	0.79	0.71	0.42	0.41	0.39	0.27	0.26	0.19	0.16	78
		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	79
		10	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	80
		12	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	81
Hollow Concrete Blocks		Gravel Aggregate										
		8	0.56	0.52	0.34	0.34	0.32	0.24	0.23	0.17	0.15	82
		12	0.49	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	83
		Cinder Aggregate										
		8	0.41	0.39	0.28	0.28	0.27	0.21	0.20	0.15	0.13	84
		12	0.38	0.36	0.26	0.26	0.25	0.20	0.19	0.15	0.13	85
		Light Weight Aggregate[e]										
		8	0.36	0.34	0.26	0.25	0.24	0.19	0.19	0.15	0.13	86
		12	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	87

[a]Based on 4 in. hard brick and remainder common brick

[b]The 8 in. and 10 in tile figures are based on two cells in the direction of heat flow. The 12 in. tile is based on three cells in the direction of heat flow. The 16 in. tile consists of one 10 in. and one 6 in. tile each having two cells in the direction of heat flow.

[c]Limestone or sandstone.

[d]These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish

[e]Expanded slag, burned clay or pumice.

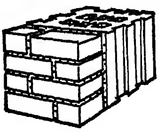
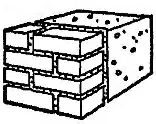
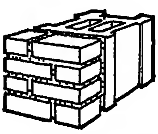
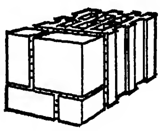
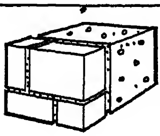
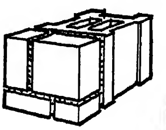
[f]Thickness of plaster assumed ¾ in.

[g]Thickness of plaster assumed ½ in

[h]Based on 2 in. furring strips; one air space.

Table 8. Coefficients of Transmission (U) of Brick and Stone Veneer Masonry Walls

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph

Typical Construction	Facing	Backing	Interior Finish (Plus Insulation Where Indicated): Plain Walls—no Interior Finish	Plaster (½ in.) on Walls	Metal Lath and Plaster[e]—Furred[g]	Gypsum Board (⅜ in.) Decorated—Furred[g]	Gypsum Lath (⅜ in.) Plastered[f]—Furred[g]	Insulating Board (½ in.) Plain or Decorated—Furred[g]	Insulating Board Lath (½ in.) Plastered[f]—Furred[g]	Insulating Board Lath (1 in.) Plastered[f]—Furred[g]	Gypsum Lath Plastered[f] Plus 1 in. Blanket Insulation—Furred[g]	Wall Number
			A	B	C	D	E	F	G	H	I	
	4 in. Brick Veneer[a]	8 in. Hollow Tile[b]	0.35	0.34	0.25	0.25	0.24	0.19	0.18	0.14	0.13	88
		8 in. Hollow Tile[b]	0.34	0.32	0.25	0.24	0.23	0.19	0.18	0.14	0.13	89
		6 in. Concrete	0.59	0.54	0.35	0.35	0.33	0.24	0.23	0.17	0.15	90
		8 in. Concrete	0.54	0.50	0.33	0.33	0.31	0.23	0.23	0.17	0.15	91
		8 in. Concrete Blocks[c] (Gravel Aggregate)	0.44	0.41	0.29	0.29	0.28	0.21	0.21	0.16	0.14	92
		8 in. Concrete Blocks[c] (Cinder Aggregate)	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	93
		8 in. Concrete Blocks[c] (Light Weight Aggregate)[d]	0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	94
	4 in. Cut Stone Veneer[a]	6 in. Hollow Tile[b]	0.37	0.35	0.26	0.26	0.25	0.19	0.19	0.15	0.13	95
		8 in. Hollow Tile[b]	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.14	0.13	96
		6 in. Concrete	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	97
		8 in. Concrete	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	98
		8 in. Concrete Blocks[c] (Gravel Aggregate)	0.47	0.44	0.30	0.30	0.29	0.22	0.21	0.16	0.14	99
		8 in. Concrete Blocks[c] (Cinder Aggregate)	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.15	0.13	100
		8 in. Concrete Blocks[c] (Light Weight Aggregate)[d]	0.32	0.30	0.23	0.23	0.22	0.18	0.17	0.14	0.12	101

[a]Calculations based on ½ in. cement mortar between backing and facing except in the case of the concrete backing which is assumed to be poured in place.

[b]The hollow tile figures are based on two air cells in the direction of heat flow.

[c]Hollow concrete blocks.

[d]Expanded slag, burned clay or pumice.

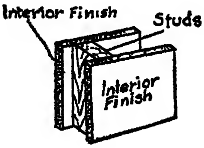
[e]Thickness of plaster assumed ¾ in.

[f]Thickness of plaster assumed ½ in.

[g]Based on 2 in. furring strips, one air space.

Table 9. Coefficients of Transmission (U) of Frame Partitions or Interior Walls[a]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides.



Interior Finish	Single Partition (Finish on one side only of studs)	Double Partition (Finish on both sides of studs): No Insulation between Studs	Double Partition (Finish on both sides of studs): 1 in. Blanket[d] between Studs. One Air Space.	Partition Number
	A	B	C	
Metal Lath and Plaster[b]	0.69	0.39	0.16	1
Gypsum Board (3/8 in.) Decorated	0.67	0.37	0.16	2
Wood Lath and Plaster	0.62	0.34	0.15	3
Gypsum Lath (3/8 in.) Plastered[c]	0.61	0.34	0.15	4
Plywood (3/8 in.) Plain or Decorated	0.59	0.33	0.15	5
Insulating Board (1/2 in.) Plain or Decorated	0.36	0.19	0.11	6
Insulating Board Lath (1/2 in.) Plastered[c]	0.35	0.18	0.11	7
Insulating Board Lath (1 in.) Plastered[c]	0.23	0.12	0.082	8

[a]Coefficients not weighted; effect of studding neglected.

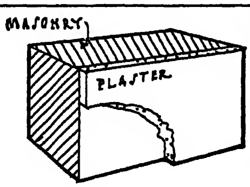
[b]Plaster assumed 3/4 in. thick.

[c]Plaster assumed 1/2 in. thick.

[d]For partitions with other insulations between studs refer to Table 6, using values in Column B of above table in left hand column of Table 6. *Example:* What is the coefficient of transmission (U) of a partition consisting of gypsum lath and plaster on both sides of studs with 2 in blanket between studs? *Solution:* According to above table, this partition with no insulation between studs (No. 4B) has a coefficient of 0.34. Referring to Table 6, it will be found that a wall having a coefficient of 0.34 with no insulation between studs, will have a coefficient of 0.10 with 2 in. of blanket insulation between studs (No 56B).

Table 10. Coefficients of Transmission (U) of Masonry Partitions

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides.



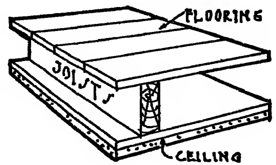
Type of Partition		Thickness of Masonry (Inches)	Type of Finish: No Finish (Plain walls)	Type of Finish: Plaster One Side	Type of Finish: Plaster Both Sides[a]	Partition Number
			A	B	C	
Hollow Clay Tile		3	0.50	0.47	0.43	9
		4	0.45	0.42	0.40	10
Hollow Gypsum Tile		3	0.35	0.33	0.32	11
		4	0.29	0.28	0.27	12
Hollow Concrete Tile or Blocks	Cinder Aggregate	3	0.50	0.47	0.43	13
		4	0.45	0.42	0.40	14
	Light Weight Aggregate[b]	3	0.41	0.39	0.37	15
		4	0.35	0.34	0.32	16
Common Brick		4	0.50	0.46	0.43	17

[a]2 in solid plaster partition, $U = 0.53$.

[b]Expanded slag, burned clay or pumice.

Table 11. Coefficients of Transmission (*U*) of Frame Construction Ceilings and Floors

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides) and are based on still air (no wind) conditions on both sides.



Type of Ceiling	Insulation Between, or on Top of, Joists (No Flooring Above)												With Flooring[g] (On Top of Ceiling Joists)		Number
	None	Insulating Board on Top of Joists		Blanket or Bat Insulation[f] Between Joists[a]			Vermiculite Insulation Between Joists[a]			Mineral Wool Insulation Between Joists[a]			Single Wood Floor[b]	Double Wood Floor[c]	
		½ In.	1 In.	1 In.	2 In.	3 In.	2 In.	3 In.	4 In.	2 In.	3 In.	4 In.			
	A	B	C	D	E	F	G	H	I	J	K	L	M	N	
No Ceiling		0.37	0.24										0.45	0.34	1
Metal Lath and Plaster[d]	0.69	0.26	0.19	0.19	0.12	0.093	0.18	0.14	0.11	0.12	0.093	0.077	0.30	0.25	2
Gypsum Board (⅜ in.) Plain or Decorated	0.67	0.26	0.18	0.19	0.12	0.092	0.18	0.13	0.10	0.12	0.092	0.077	0.30	0.24	3
Wood Lath and Plaster	0.62	0.25	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.24	4
Gypsum Lath (⅜ in.) Plastered[e]	0.61	0.25	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.24	5
Plywood (⅜ in.) Plain or Decorated	0.59	0.24	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.23	6
Insulating Board (½ in.) Plain or Decorated	0.36	0.19	0.15	0.16	0.10	0.082	0.14	0.12	0.097	0.10	0.082	0.069	0.22[h]	0.19[h]	7
Insulating Board Lath (½ in.) Plastered[e]	0.35	0.19	0.15	0.15	0.10	0.081	0.14	0.11	0.096	0.10	0.081	0.068	0.21	0.18	8
Insulating Board Lath (1 in.) Plastered[e]	0.23	0.15	0.12	0.12	0.089	0.072	0.12	0.097	0.084	0.089	0.072	0.061	0.16	0.14	9

[a]Coefficients corrected for framing on basis of 15 per cent area, 2 in. x 4 in. (nominal) framing, 16 in. on centers.

[b]²⁵⁄₃₂ in. yellow pine or fir.

[c]²⁵⁄₃₂ in. pine or fir sub-flooring plus ¹³⁄₁₆ in. hardwood finish flooring.

[d]Plaster assumed ¾ in. thick.

[e]Plaster assumed ½ in. thick.

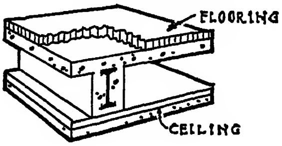
[f]Based on insulation in contact with ceiling and consequently no air space between.

[g]For coefficients for constructions in Columns M and N (except No. 1) with insulation between joists, refer to Table 6. *Example:* The coefficient for No. 3-N of Table 11 is 0.24. With 2 in. blanket insulation between joists, the coefficient will be 0.093. (See Table 6.) (Column D of Table 6 applicable only for 3⅝ in. joists.)

[h]For ²⁵⁄₃₂ in. insulating board sheathing applied to the under side of the joists, the coefficient for *single* wood floor (Column M) is 0.18 and for *double* wood floor (Column N) is 0.16. For coefficients with insulation between joists, see Table 6

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides.



TYPE OF CEILING	THICKNESS OF CONCRETE[g] (INCHES)	TYPE OF FLOORING					NUMBER
		No Flooring (Concrete Bare)	Tile[a] or Terrazzo Flooring on Concrete	⅛ In. Asphalt Tile[b] Directly on Concrete	Parquet[c] Flooring In Mastic on Concrete	Double Wood Floor on Sleepers[d]	
		A	B	C	D	E	
No Ceiling	3	0 68	0 65	0 66	0.45	0.25	1
	6	0.59	0 56	0 58	0.41	0.23	2
	10	0.50	0.48	0 49	0.36	0.22	3
½ in. Plaster Applied to Underside of Concrete	3	0.62	0.59	0 60	0.43	0.24	4
	6	0.54	0 52	0 53	0.39	0.22	5
	10	0.46	0.44	0 45	0.34	0.21	6
Metal Lath and Plaster[e]—Suspended or Furred	3	0.38	0 37	0 37	0.30	0.19	7
	6	0.35	0 34	0 35	0.28	0.18	8
	10	0.32	0 31	0 32	0.26	0 17	9
Gypsum Board (⅜ in.) and Plaster[f]—Suspended or Furred	3	0.36	0.35	0 35	0.28	0.19	10
	6	0.33	0.32	0 33	0.27	0 18	11
	10	0.30	0.29	0 30	0.24	0.17	12
Insulating Board Lath (½ in.) and Plaster[f] Suspended or Furred	3	0 25	0 24	0 25	0.21	0.15	13
	6	0.23	0.23	0.23	0.20	0.15	14
	10	0.22	0.21	0 22	0.19	0.14	15

[a]Thickness of tile assumed to be 1 in.

[b]Conductivity of Asphalt Tile assumed to be 3 1

[c]Thickness of wood assumed to be 13/16 in ; thickness of mastic, ⅛ in. (k = 4.5). Col. D may also be used for concrete covered with carpet

[d]Based on 25/32 in yellow pine or fir sub-flooring and 13/16 in. hardwood finish flooring with an air space between sub-floor and concrete

[e]Thickness of plaster assumed to be ¾ in.

[f]Thickness of plaster assumed to be ½ in.

[g]For other thicknesses of concrete, interpolate.

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING

U = 0.10[a] Btu per (hour) (square foot) (Fahrenheit degree temperature difference between the ground and the air over the floor).

[a]Until more complete data are available, it is recommended that a coefficient of 0.10 be used for all types of concrete floors on the ground, with or without insulation. For basement wall below grade, use the same average coefficient (0 10) A lower ground temperature should, however, be used for walls than floors as explained in Chapter 14 For further data see A S H V.E. RESEARCH REPORT No 1213—Heat Loss Through Basement Walls and Floors, by F C. Houghten, S. I Taimuty, Carl Gutberlet and C. J. Brown (A.S H.V E. TRANSACTIONS, Vol. 48, 1942, p. 369).

exact compliance with specifications, and a factor of safety may be employed as a precaution when it is judged desirable.

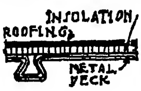
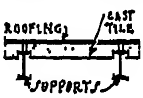
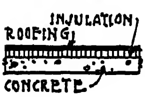
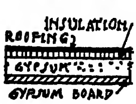
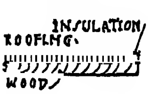
Roof Coefficients

Computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Values for roofs containing Spanish and French clay roofing tile are assumed the same as for slate roofs. Values for pitched roofs in Table 16 apply where the roof is over a heated attic or top floor so that the heat passes directly through the roof structure, including any interior finish material.

TABLE 14. COEFFICIENTS OF TRANSMISSION (*U*) OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING. NO CEILING—UNDER SIDE OF ROOF EXPOSED

(See Table 15 for Flat Roofs with Ceilings)

These coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

TYPE OF ROOF DECK	THICKNESS OF ROOF DECK (INCHES)	NO INSULATION	INSULATION ON TOP OF DECK (COVERED WITH BUILT-UP ROOFING)							NUMBER
			INSULATING BOARD (Thickness Below)				CORKBOARD (Thickness Below)			
			½ In.	1 In.	1½ In.	2 In.	1 In.	1½ In.	2 In.	
		A	B	C	D	E	F	G	H	
Flat Metal Roof Deck[a] 		1.06	0.39	0.24	0.18	0.14	0.23	0.17	0.13	1
Precast Cement Tile 	1⅝ in.	0.84	0.37	0.24	0.17	0.14	0.22	0.16	0.13	2
Concrete 	2 in. 4 in. 6 in.	0.82 0.72 0.65	0.36 0.34 0.33	0.24 0.23 0.22	0.17 0.17 0.16	0.14 0.13 0.13	0.22 0.21 0.21	0.16 0.16 0.15	0.13 0.12 0.12	3 4 5
Gypsum Fiber Concrete[b] on ½ in. Gypsum Board 	2½ in. 3½ in.	0.38 0.31	0.24 0.21	0.18 0.16	0.14 0.13	0.12 0.11	0.17 0.15	0.13 0.12	0.11 0.10	6 7
Wood[c] 	1 in. 1½ in. 2 in. 3 in.	0.49 0.37 0.32 0.23	0.28 0.24 0.22 0.17	0.20 0.17 0.16 0.14	0.15 0.14 0.13 0.11	0.12 0.11 0.11 0.096	0.19 0.17 0.16 0.13	0.14 0.13 0.12 0.11	0.12 0.11 0.10 0.091	8 9 10 11

[a]Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per (hour)(square foot of projected area)(Fahrenheit degree difference in temperature) based on an outside wind velocity of 15 mph.

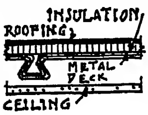
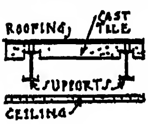
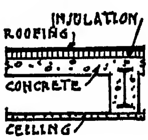
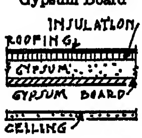
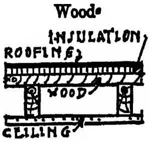
[b]87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ½ in. gypsum board.

[c]Nominal thicknesses specified—actual thicknesses used in calculations

TABLE 15 COEFFICIENTS OF TRANSMISSION (U) OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING. WITH LATH AND PLASTER CEILINGS[a]

(See Table 14 for Flat Roofs with No Ceilings)

These coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

TYPE OF ROOF DECK	THICKNESS OF ROOF DECK (INCHES)	NO INSULATION	INSULATION ON TOP OF DECK (COVERED WITH BUILT-UP ROOFING)							NUMBER
			INSULATING BOARD (Thickness Below)				CORKBOARD (Thickness Below)			
			½ In.	1 In.	1½ In.	2 In.	1 In.	1½ In.	2 In.	
		A	B	C	D	E	F	G	H	
Flat Metal Roof Deck 		0.46	0 27	0.19	0.15	0 12	0.18	0 14	0.11	12
Precast Cement Tile 	1⅝ in.	0.43	0.26	0.19	0 15	0.12	0.18	0 14	0.11	13
Concrete 	2 in.	0 42	0.26	0.19	0 14	0 12	0 18	0.14	0.11	14
	4 in.	0.40	0.25	0 18	0 14	0.12	0 17	0.13	0 11	15
	6 in.	0.37	0.24	0 18	0.14	0 11	0 17	0.13	0.11	16
Gypsum Fiber Concrete[b] on ½ in. Gypsum Board 	2½ in.	0.27	0.19	0.15	0 12	0 10	0.14	0.12	0 097	17
	3½ in.	0.23	0 17	0.14	0.11	0 097	0.13	0 11	0 091	18
Wood[c] 	1 in.	0 31	0 21	0 16	0.13	0.11	0.15	0 12	0.10	19
	1½ in.	0.26	0 19	0.15	0.12	0 10	0.14	0.11	0.095	20
	2 in.	0 24	0.17	0.14	0 11	0 097	0.13	0.11	0 092	21
	3 in.	0 18	0.14	0.12	0 10	0.087	0.11	0 095	0.082	22

[a]Calculations based on metal lath and plaster ceilings, but coefficients may be used with sufficient accuracy for gypsum lath or wood lath and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

[b]87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ½ in. gypsum board.

[c]Nominal thicknesses specified—actual thicknesses used in calculations.

Table 16. Coefficients of Transmission (U) of Pitched Roofs

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

Type of Ceiling (Applied Directly to Roof Rafters)	Wood Shingles (On 1 x 4 Wood Strips[c] Spaced 2 In. Apart)				Asphalt Shingles or Roll Roofing (On Solid Wood Sheathing)[c]				Slate or Tile[b] (On Solid Wood Sheathing)[c]				Number
	Insulation Between Rafters				Insulation Between Rafters				Insulation Between Rafters				
	None	Blanket or Bat (Thickness Below)			None	Blanket or Bat (Thickness Below)			None	Blanket or Bat (Thickness Below)			
		1 In.	2 In.	3 In.		1 In.	2 In.	3 In.		1 In.	2 In.	3 In.	
	A	B	C[a]	D[a]	E	F	G[a]	H[a]	I	J	K[a]	L[a]	
No Ceiling Applied to Rafters	0.48[f]	0.15	0.10	0.081	0.52[f]	0.15	0.11	0.084	0.55[f]	0.16	0.11	0.085	1
Metal Lath and Plaster[d]	0.31	0.14	0.10	0.081	0.33	0.15	0.10	0.083	0.34	0.15	0.10	0.083	2
Gypsum Board (⅜ in.) Decorated	0.30	0.14	0.10	0.080	0.32	0.15	0.10	0.082	0.33	0.15	0.10	0.083	3
Wood Lath and Plaster	0.29	0.14	0.10	0.080	0.31	0.14	0.10	0.081	0.32	0.15	0.10	0.082	4
Gypsum Lath (⅜ in.) Plastered[e]	0.29	0.14	0.10	0.079	0.31	0.14	0.10	0.081	0.32	0.15	0.10	0.082	5
Plywood (⅜ in.) Plain or Decorated	0.29	0.14	0.099	0.079	0.30	0.14	0.10	0.081	0.31	0.15	0.10	0.081	6
Insulating Board (½ in.) Plain or Decorated	0.22	0.12	0.090	0.072	0.23	0.12	0.091	0.074	0.24	0.13	0.092	0.074	7
Insulating Board Lath (½ in.) Plastered[e]	0.22	0.12	0.088	0.072	0.22	0.12	0.090	0.073	0.23	0.12	0.091	0.074	8
Insulating Board Lath (1 in.) Plastered[e]	0.16	0.10	0.078	0.064	0.17	0.10	0.079	0.065	0.17	0.10	0.080	0.066	9

[a]Coefficients corrected for framing on basis of 15 per cent area, 2 in. x 4 in. (nominal), 16 in. on centers.
[b]Figures in Columns I, J, K and L may be used with sufficient accuracy for rigid asbestos shingles on wood sheathing. Layer of slater's felt neglected.
[c]Sheathing and wood strips assumed 25/32 in. thick.
[d]Plaster assumed ¾ in. thick.
[e]Plaster assumed ½ in. thick.
[f]No air space included in 1-A, 1-E or 1-I; all other coefficients based on one air space.

Table 17. Combined Coefficients of Transmission (U) of Pitched Roofs[a] and Horizontal Ceilings—Based on Ceiling Area[b]

Coefficients are expressed in Btu per (hour) (square foot of ceiling area) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

Ceiling Coefficient[f] (From Table 11)	Type of Roofing and Roof Sheathing: Wood Shingles on Wood Strips[d]			Asphalt Shingles[c] or Roll Roofing on Wood Sheathing[e]			Number
	No Roof Insulation (Rafters Exposed) ($U_r = 0.48$)	½ In. Insulating Board on Under Side of Rafters ($U_r = 0.22$)	1 In. Insulating Board on Under Side of Rafters ($U_r = 0.16$)	No Roof Insulation (Rafters Exposed) ($U_r = 0.53$)	½ In. Insulating Board on Under Side of Rafters ($U_r = 0.23$)	1 In. Insulating Board on Under Side of Rafters ($U_r = 0.17$)	
	A	B	C	D	E	F	
0.10	0.085	0.073	0.066	0.087	0.074	0.067	19
0.11	0.092	0.078	0.07	0.094	0.079	0.071	20
0.12	0.099	0.082	0.074	0.10	0.083	0.075	21
0.13	0.11	0.087	0.078	0.11	0.088	0.079	22
0.14	0.11	0.091	0.081	0.11	0.093	0.083	23
0.15	0.12	0.096	0.084	0.12	0.097	0.086	24
0.16	0.13	0.10	0.087	0.13	0.10	0.089	25
0.17	0.13	0.10	0.090	0.13	0.10	0.092	26
0.18	0.14	0.11	0.093	0.14	0.11	0.095	27
0.19	0.14	0.11	0.095	0.15	0.11	0.098	28
0.20	0.15	0.11	0.098	0.15	0.12	0.10	29
0.21	0.15	0.12	0.10	0.16	0.12	0.10	30
0.22	0.16	0.12	0.10	0.17	0.12	0.11	31
0.23	0.16	0.12	0.10	0.17	0.12	0.11	32
0.24	0.17	0.13	0.11	0.18	0.12	0.11	33
0.25	0.17	0.13	0.11	0.18	0.13	0.11	34
0.26	0.18	0.13	0.11	0.19	0.13	0.11	35
0.27	0.18	0.13	0.11	0.19	0.13	0.12	36
0.28	0.19	0.14	0.12	0.19	0.14	0.12	37
0.29	0.19	0.14	0.12	0.20	0.14	0.12	38
0.30	0.20	0.14	0.12	0.20	0.14	0.12	39
0.34	0.21	0.15	0.12	0.22	0.15	0.13	40
0.35	0.22	0.15	0.13	0.22	0.15	0.13	41
0.36	0.22	0.15	0.13	0.23	0.15	0.13	42
0.37	0.23	0.15	0.13	0.23	0.16	0.13	43
0.45	0.25	0.17	0.13	0.26	0.17	0.14	44
0.59	0.29	0.18	0.14	0.30	0.19	0.15	45
0.61	0.29	0.18	0.15	0.31	0.19	0.15	46
0.62	0.30	0.19	0.15	0.31	0.19	0.15	47
0.67	0.31	0.19	0.15	0.33	0.20	0.16	48
0.69	0.31	0.19	0.15	0.33	0.20	0.16	49

[a]Calculations based on ⅓ pitch roof ($n = 1.2$) using the following formula:

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}}$$

U = combined coefficient to be used with ceiling area.
U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling.

[b]Use ceiling area (not roof area) with these coefficients.

[c]Coefficients in Columns D, E and F may be used with sufficient accuracy for tile, slate and rigid asbestos shingles on wood sheathing.

[d]Based on 1 x 4 in. strips spaced 2 in. apart.

[e]Sheathing assumed 25/32 in. thick.

[f]Values of U_{ce} to be used in this column may be selected from Table 11.

TABLE 18. COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS, SKYLIGHTS AND GLASS BLOCK WALLS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in the temperature between the air inside and outside of the door, window, skylight or wall) and are based on an outside wind velocity of 15 mph.

Section A. Windows and Skylights		SINGLE	DOUBLE	TRIPLE
	U	1.13[ac]	0.45[ae]	0.281[ae]

Section B. Solid Wood Doors[bc]	NOMINAL THICKNESS INCHES	ACTUAL THICKNESS INCHES	U EXPOSED DOOR	U[d] WITH GLASS STORM DOOR
	1	25/32	0.69	0.42
	1¼	1 1/16	0.59	0.38
	1½	1 5/16	0.52	0.35
	1¾	1⅜	0.51	0.35
	2	1⅝	0.46	0.32
	2½	2⅛	0.38	0.28
	3	2⅝	0.33	0.25

Section C. Hollow Glass Block Walls	DESCRIPTION	U STILL AIR BOTH SIDES	U STILL AIR INSIDE 15 MPH OUTSIDE
	Smooth surface glass blocks 7¾ x 7¾ x 3⅞ in. thick	0.40	0.49
	Ribbed surface glass blocks 7¾ x 7¾ x 3⅞ in. thick	0.38	0.46

[a]See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.

[b]Computed using $C = 1.15$ for wood; $f_i = 1.65$ and $f_o = 6.0$.

[c]It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1.13 Btu per (hour) (square foot) (degree difference between inside and outside air temperatures).

[d]These values may also be used with sufficient accuracy for wood storm doors. Neglect storm doors if loose and use values for exposed doors.

[e]Air spaces assumed to be ¾ in. or more in width.

Combined Ceiling and Roof Coefficients

If the attic space between ceiling and roof is unheated, the combined coefficient from room air below the ceiling to exterior air can be calculated from the following formula.

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}} \tag{5}$$

where

U = combined coefficient to be used with ceiling area.

U_r = coefficient of transmission of roof.

U_{ce} = coefficient of transmission of ceiling.

n = ratio of roof area to ceiling area.

It should be noted that the over-all coefficient U should be multiplied by the ceiling area to determine heat loss and not by the roof area. Values of U_r and U_{ce} should be calculated using a value of 2.2 (the reciprocal of one-half the air space resistance) rather than 1.65 for the conductances of surfaces facing the attic, since the attic is equivalent to an air space.

If the attic contains windows, dormers and vertical wall spaces and if their area is small compared to that of the roof, they may be considered

part of the roof area. For accuracy, the sum of the coefficients of each individual section multiplied by its percentage of the total area should be used as U_r. Where large vertical wall areas in the attic are involved, it is preferable to estimate the attic temperature as illustrated in Chapter 14 and calculate the heat loss through the ceiling by multiplying the value of U_{ce} for the ceiling by the difference in temperature above and below the ceiling.

Basement Floor, Basement Wall and Concrete Slab Floor Coefficients

The heat transfer through basement walls and floors to the ground is dependent on the temperature difference between the air within and that of the ground, on the material constituting the wall or floor, and on the conductivity of the surrounding earth. The conductivity of the earth will vary with local conditions and is usually unknown. Tests [8] at the A.S.H.V.E. Research Laboratory indicate a heat flow of approximately 2.0 Btu per (hour) (square foot) through an uninsulated concrete basement floor with a temperature difference of 20 F between ground temperature and the air temperature 6 in. above the floor. Based on this result, a coefficient of 0.10 Btu per (hour) (square foot) (Fahrenheit degree temperature difference) is recommended for calculation where it is desirable to allow for the small basement floor heat loss, *e.g.* for heated basements.

For basement walls the same coefficient may be used, but due to closer proximity to the surface of the ground, the temperature difference for winter design conditions will be greater than for the floor. The test results indicate a unit area heat loss, at mid-height of the basement wall, approximately twice that of the same floor area.

For concrete slab floors laid in contact with the ground at grade level, recent tests [9] indicate that for small floor areas (equal to that of a house 25 ft square) the heat loss may be calculated as proportional to the length of exposed edge rather than total area. This amounts to 0.81 Btu per (hour) (lineal foot of exposed edge) (Fahrenheit degree difference between the inside air temperature and the average outside air temperature). It should be noted that this may be appreciably reduced by insulating the edges of the floor from the abutting wall.

CONDENSATION IN BUILDINGS

Water vapor in the air within a building condenses if it comes in contact with surfaces at or below its dew-point temperature. It also will be transmitted into or through a wall, floor or ceiling, if a vapor pressure difference exists between the opposite sides, at a rate determined by the permeability of the materials encountered [10] (see Table 17 Chapter 15). Building practice must take account of these facts in avoiding (1) surface condensation on interior building surfaces (walls, ceilings, roofs or glass) and (2) interstitial condensation or accumulation of condensation in the voids within the structure. The conditions under which surface condensation will take place are directly dependent on surface temperature and upon the relative humidity of the air in contact. Limiting maximum relative humidities for walls, roofs or glass having transmission coefficients up to 1.2 Btu for outside temperatures from —30 F to 40 F and for 70 F inside temperature may be obtained from Fig. 4 [11].

Surface condensation may be controlled by air conditioning, ventilation for the purpose of removing water vapor, particularly from laundries

and from kitchens during lengthy periods of cooking, elimination of sources of vapor such as unvented gas stoves, or by local application of heat or insulation to raise surface temperatures.

Interstitial condensation results from vapor transmission which is dependent on vapor pressure differential and the ratio of the rate at which vapor may enter the materials of the structure to that at which it passes out of the structure. The proper installation of a vapor barrier on the warm side of the structure, where the higher vapor pressure usually exists, will reduce greatly the amount of vapor entering the building construction and will minimize the possibility of objectionable condensation. Such a barrier may consist of a coated paper applied

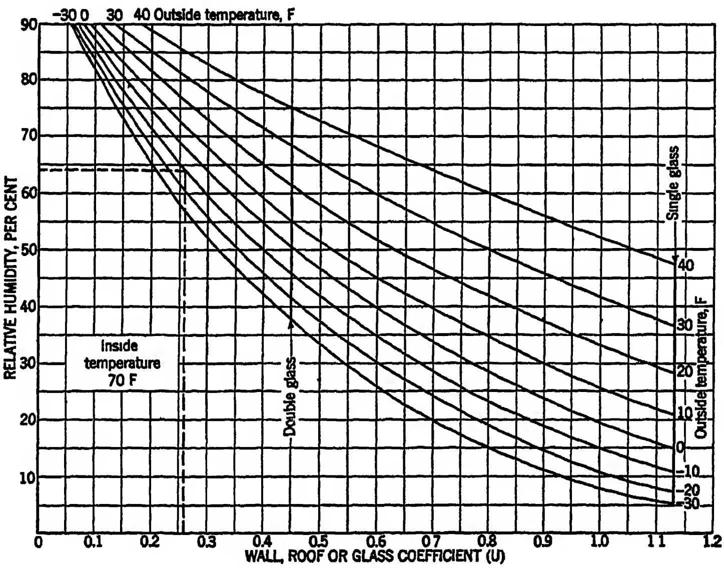


Fig. 4. Permissible Relative Humidities for Various Transmission Coefficients

under the plaster, a coated plaster base, or a vapor-resistant finish applied to the interior wall surface [12]. A vapor barrier is tentatively defined as a material having a water vapor permeability not exceeding 1.25 grains per (hour) (square foot) (inch of Hg vapor pressure differential).

To prevent condensation on the under side of roofs above attic spaces over insulated ceilings, ventilation with outside air may be provided through suitable louvers or other roof ventilators.

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CHAPTER 7

Performance of Air Heating and Cooling Coils

Performance of Heating and Dry Cooling Coils, Over-all Coefficient of Heat Transfer, Performance of Dehumidifying Coils, External Film Coefficient, Internal Film Coefficient, Determining Size of Cooling Coil

THE surfaces described in this chapter are for heating or cooling an air stream under the conditions of forced convection. Such surfaces may be made up of a number of banks of tubes assembled in the field or the entire assembly may be factory constructed. They may be made of either bare or finned tubing, but regardless of their construction they are generally referred to as *coils*. A description of the types, application, and selection of such coils is given in Chapter 25. Therefore, this chapter is confined to the theoretical considerations which affect the calculation of the performance of these coils.

PERFORMANCE OF HEATING AND DRY COOLING COILS

The performance of heating and dry cooling coils depends in general upon:

1. The over-all coefficient of heat transfer from the fluid within the coil to the air it heats or cools.

2. The mean temperature difference between the fluid within the coil and the air flowing over the coil.

3. The physical dimensions of the coil.

Thus, for any one definite operating condition, the heating or cooling capacity of a given coil is expressed by the following basic formula:

$$q_t = U \times (MTD) \times A \times N \qquad (1)$$

where

q_t = total heat transferred by the coil, Btu per (hour) (square foot of coil face area).

U = over-all coefficient of heat transfer, Btu per (hour) (square foot of external coil surface) (Fahrenheit degree temperature difference between the fluid within the coil and the air flowing over the coil).

MTD = mean temperature difference, Fahrenheit degrees, between the fluid within the coil and the air passing over it. (This is commonly taken as the logarithmic mean temperature difference.)

A = external surface area of the given coil, square feet per (square foot of coil face area) (row of coil depth).

N = number of rows of coil depth.

Over-all Coefficient of Heat Transfer

Of all factors affecting the performance of heating or dry cooling coils, the over-all coefficient of heat transfer is the most difficult to determine as it is influenced by several factors which depend upon coil design and conditions of operation.

Considering any coil, whether of bare pipe or of finned type, the over-all heat transfer coefficient for a given size and design of coil can always be

considered as a combined effect of three individual heat transfer coefficients, namely:

1. The film coefficient of heat transfer between air and the external surface of the coil, usually given in Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference).

2. The coefficient of heat transfer through the coil material—tube wall, fins, ribs, etc.

3. The film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, usually given in Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature difference).

These three individual coefficients acting in series result in an over-all coefficient of heat transfer in accordance with the basic laws given in Chapters 5 and 6. For a *bare pipe* coil the over-all coefficient of heat transfer, whether for heating or for cooling (without dehumidification), can be expressed by a simplified basic formula as follows:

$$U = \frac{1}{\frac{R}{h_r} + \frac{L}{k} + \frac{1}{h_a}} \tag{2}$$

where

U = over-all coefficient of heat transfer, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and fluid within the coil).

h_r = film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature difference between that surface and the average fluid temperature).

h_a = film coefficient of heat transfer between air and the external surface of the coil, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between the mass of air and the external surface).

k = conductivity of material from which the bare pipe is constructed, Btu per (hour) (square foot) (Fahrenheit degree per inch thickness).

L = thickness of tube wall, inches.

R = ratio between external and internal surface of the bare tube, usually varying from 1.03 to 1.15 for the tube used in typical heating or cooling coils. This ratio R is inserted in the formula in order to place internal fluid coefficient of heat transfer on the basis of external surface.

Frequently, when pipe or tube walls are thin and of material having high conductivity (as is the case in construction of typical heating and cooling coils) the term L/k in Equation 2 becomes negligible and is generally disregarded. (The effect of the term L/k in typical bare pipe heating or cooling coils seldom exceeds 1 to 2 per cent of the over-all coefficient). Thus, in its simplest form, for bare pipe:

$$U = \frac{1}{\frac{R}{h_r} + \frac{1}{h_a}} \tag{3}$$

For finned coils the formula[1] for the over-all coefficient of heat transfer can be conveniently written:

$$U = \frac{1}{\frac{R}{h_r} + \frac{1}{\eta h_a}} \tag{4}$$

in which the term η, called the *fin efficiency*, is introduced to allow for the resistance to heat flow encountered in the fins.

The term R, in this case, is the ratio of *total* external surface to internal surface. For typical designs of finned coils for heating or cooling, this ratio varies from 10 to 30. Term R is again introduced to place the internal surface coefficient of heat transfer on a basis of external surface.

In the discussions which follow, coefficients h_r and ηh_a will be considered separately, and also various ways of combining them will be outlined.

The performances of all heating and dry cooling coils are influenced by these same factors. But, when cooling coils operate wet or act as dehumidifying coils, the performance cannot be predicted on the basis of over-all coefficients and an analysis must be made on the basis of individual film coefficients as will be explained.

PERFORMANCE OF DEHUMIDIFYING COILS

When a cooling coil operates with a surface temperature which is below the dew-point of the air entering the coil, moisture is condensed and the air leaves the coil with a humidity ratio lower than it had when

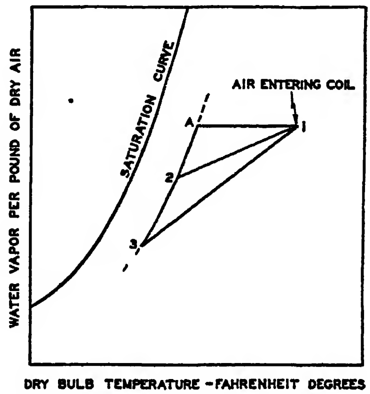

FIG. 1. PERFORMANCE OF DEHUMIDIFYING COIL

it entered the coil. To understand the performance of surface coils under such conditions, assume that air enters a cooling coil at conditions corresponding to point 1 in Fig. 1. As long as the surface temperature of the coil is above the dew-point, the air is cooled without dehumidification, and its condition leaving the coil will be somewhere on line 1-A. Its exact position on this line depends on the air velocity and the external film coefficient as well as upon the surface temperature. When the surface temperature just equals the dew-point, the air leaves with conditions represented by point A. If the surface temperature is below the dew-point, condensation takes place, and the air has a final condition somewhere along the line A-2-3 which is a line at a constant horizontal distance from the saturation curve. It should be understood that the line 1-A-2-3 is not intended to represent the path of the condition of the air as it passes through the coil from row to row. It is simply the path traced by the exit air conditions as the surface temperature is gradually reduced with other conditions remaining constant [2].

In the process of dehumidification, since heat is being transferred to the coil surface by two different mechanisms, (convection and condensation), it is evident that an over-all coefficient of heat transfer cannot be determined by the same method used for heating and for dry cooling coils. However, if it is assumed that the sensible heat transfer of a dehumidifying coil is unaffected by the presence of moisture on its surface, Equation 5 may be obtained to express this part of the heat

transfer in terms of the external film coefficient and the surface temperature.

$$q_s = h_a \times A \times N \times (MTD_a) \tag{5}$$

where

q_s = sensible heat transferred, Btu per (hour) (square foot of coil face area).
t_1 = dry-bulb temperature of air entering coil, Fahrenheit degrees.
t_2 = dry-bulb temperature of air leaving coil, Fahrenheit degrees.
t_s = average temperature of coil external surface, Fahrenheit degrees.
MTD_a = logarithmic mean temperature difference between air and coil surface = $\dfrac{t_1 - t_2}{\log_e \dfrac{t_1 - t_s}{t_2 - t_s}}$

If Equation 5 is combined with another equation expressing sensible heat transfer in terms of mass velocity and temperature difference, the variables may be arranged in the following form (which is useful for the solution of dehumidification problems and for the determination of h_a from test data):

$$\frac{h_a\, A\, N\, (t_1 - t_2)}{\log_e \dfrac{t_1 - t_s}{t_2 - t_s}} = 0.243\, G\, (t_1 - t_2)$$

or,

$$\frac{h_a\, A\, N}{0.243G} = \log_e \frac{t_1 - t_s}{t_2 - t_s} \tag{6}$$

where

0.243 = specific heat of humid air, Btu per (pound) (Fahrenheit degree).
G = air mass velocity, pounds per (hour) (square foot of coil face area).

An examination of Fig. 1 will reveal that when t_s is at the dew-point of the entering air:

$$\frac{t_1 - t_s}{t_a - t_s} = \frac{t_1 - t_{dp1}}{t_a - t_{dp1}}$$

and when t_s is below the dew-point:

$$\frac{t_1 - t_s}{t_2 - t_s} = \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}}$$

Therefore, Equation 6 may be written in its most useful form as:

$$\frac{h_a\, A\, N}{0.243G} = \log_e \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = \log_e \frac{t_1 - t_s}{t_2 - t_s} \tag{7}$$

where

t_a = minimum dry-bulb possible without dehumidification, Fahrenheit degrees.
t_{dp1} = dew-point of air entering coil, Fahrenheit degrees.
t_{dp2} = dew-point of air leaving coil, Fahrenheit degrees.

This equation may be used to establish a line as A-2-3 for a given coil if h_a is known for the coil, or it *may be used to determine h_a from test data for the purpose of rating coils.* The use of this equation for coil selection is illustrated in Example 1 at the end of the chapter. Equation 7 is also important as a means of determining the external film coefficient.

External Film Coefficient

While formulas have been developed expressing the film coefficient h_a for air passing parallel to a plane surface, they cannot be used directly

for fins on tubes because of air turbulence and because of the temperature gradient prevalent from the edge of a fin to its center. It is therefore necessary to make tests to evaluate the combined term ηh_a. The term, ηh_a, will be written merely h_a in this discussion as there is no necessity for separately evaluating η and because values of h_a are usually applied only to the particular coils for which tests are made.

The air side coefficient, h_a, of a coil of particular dimensions is an exponential function of the mass velocity of the air:

$$h_a = Z G^n \tag{8}$$

where

- h_a = film coefficient of heat transfer, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and average surface temperature).
- G = air mass velocity, pounds per (hour) (square foot of coil face area).
- Z and n = constants which depend upon both air turbulence and surface arrangement.

Evaluation of constants Z and n may be accomplished through the use of test data in Equation 7 which gives values of h_a directly from the results of any *wet coil test.* If h_a, calculated in this manner, is plotted against values of G which prevailed during the tests a straight line should result on logarithmic coordinates. The slope of this line is the value of n. The value of Z may then be determined by direct substitution in Equation 8.

For finned coils of different designs, values of Z and n are extremely variable, depending on the particular design and arrangement of the coil surface. Therefore, it is desirable that these constants be determined directly from test data for each type of coil surface.

Internal Film Coefficient

The internal film coefficient, h_r which appears in Equation 3, is evaluated in various ways, depending upon the nature of the fluid, and whether the fluid is changing state.

When evaporating refrigerants are used in tubes, the temperature of the fluid is fairly constant, being affected principally by pressure drop through the tubes, by superheat of the evaporated refrigerant, and by the presence of oil in solution. To obtain maximum coil capacity it is necessary to keep the pressure drop through the tubes at a minimum, to keep the superheat as low as possible without carrying liquid back to the compressor, and to arrange for good separation and return of oil to the compressor. Another important factor is the removal of gas to keep the tube surface flooded with liquids as much as possible. The internal film coefficient is markedly increased by heavy heat loads, because the increased turbulence and gas velocity cause good contact of the liquid with the tubes. Values of h_r usually lie between 150 and 450. For rating of dehumidifying coils, satisfactory results are obtainable by first determining the average external surface temperature from Equation 7, and then using the difference between the external film temperature and the refrigerant for evaluating h_r in Equation 9.

$$h_r = \frac{q_t}{\frac{A\,N}{R}\,(t_s - t_r)} \tag{9}$$

where

h_r = internal film coefficient of heat transfer, Btu per (hour) (square foot of internal tube surface) (Fahrenheit degree).

t_r = average refrigerant temperature, Fahrenheit degrees.

The term $(t_s - t_r)$ is commonly written Δt. To evaluate h_r by this method the same tests that were required to determine h_a may be used.

When water is the cooling medium in tubes, the rate of heat transfer is a function of its velocity, which influences the number of contacts of the water molecules with the tube surface, per unit of time. Increased water velocity and reduced tube diameter cause increased heat transfer. Heat transfer is also greater at higher temperatures of the water. The basic formula for the film coefficient of heat transfer for flow of water in smooth tubes is as follows:

$$h_r = 1.5\,(t + 100)\,\frac{V^{0.8}}{D^{0.2}} \qquad (10)$$

where

V = water velocity, feet per second.

D = internal diameter of tube, inches.

t = average water temperature, Fahrenheit degrees.

Equation 10 should not be used when Reynolds Number is less than 2000.

Since, in the case of finned tubes using water as a refrigerant, test values of h_r based on the calculated surface temperature for the entire coil may be lower than those obtained by use of Equation 10, actual test results are preferred if available.

When saturated steam is condensed in the tubes of coils, the film coefficient h_r varies from 1000 to 2000, depending on freedom from air in the steam, and upon good drainage of the tubes. The coefficient is fairly constant for a particular coil, giving values of Δt that are directly proportional to q_t. However, if water coil test results are analyzed on a row-by-row basis good agreement with Equation 10 will result.[3]

The use of turbulence promoters increases the value of h_r for liquids in tubes at the expense of pressure drop. The increase obtained depends upon the type of turbulence promoter and the rate of flow. No general statement can be made regarding their use and it is best to refer to detailed papers on this subject for further information.[3,4]

Determining Size of Cooling Coil

To illustrate the use of individual film coefficients in coil calculations, the procedure for selecting the proper size cooling coil and for determining exit air condition, coil surface temperature, total coil load and refrigerant temperature is outlined in Example 1.

Example 1. An industrial application requires the cooling of a certain quantity of air from a condition of 102 F dry-bulb and 85 F wet-bulb to a final condition of 80.5 F dry-bulb and 73 F wet-bulb. The air velocity across the coil is to be 400 fpm and coil data are as follows: h_a = 10.7 at 400 fpm, h_r = 325, external surface area = 15 sq ft per (square foot of face) (row of coil depth), ratio of external surface area to internal surface area = 15.

Solution. (1) Lay out the problem psychrometry as indicated in Fig. 2 and note that the minimum horizontal distance between the load ratio line and the saturation curve is 1.8 F dry-bulb at point A Fig. 2. This means that $t_2 - t_{dp2}$ in Equation 7 must not be less than 1.8. Therefore, Equation 7 should be solved for N to determine the proper number of rows to be used for the coil.

$$\frac{h_a\,A\,N}{0.243\,G} = \log_e \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = \log_e \frac{102 - 80}{1.8} = \log_e 12.22 = 2.5$$

Then substituting values for h_a, A, and G, N may be found as follows:

$$\frac{10.7 \times 15N}{0.243 \times 1740} = 2.5 \text{ from which, } N = 6.58$$

(2) This establishes the maximum whole number of coil rows that can be used as 6 and it is now possible to determine the actual location of the exit air conditions from Equation 7 by solving for the actual value of $t_2 - t_{dp2}$ for a 6 row coil.

$$\frac{10.7 \times 15 \times 6}{0.243 \times 1740} = \log_e \frac{102 - 80}{t_1 - t_{dp2}} = 2.275$$

This establishes values of 9.78 for $\dfrac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = R$ and 2.25 for $t_2 - t_{dp2}$.

(3) Next, the exit air condition at 57.3 F dry-bulb and 56 F wet-bulb as shown at B, is found by locating a point on the load ratio line at a horizontal distance of 2.25 dry-bulb degrees from the saturation curve.

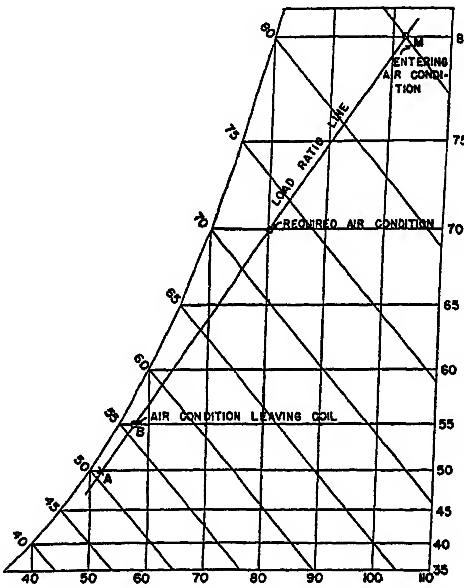

FIG. 2. PSYCHROMETRIC LAYOUT FOR COIL SELECTION

(4) The surface temperature may now be found from Equation 7 which may also be written as:

$$t_s = \frac{Bt_2 - t_1}{B - 1}$$

where

$$B = \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}}$$

$$t_s = \frac{9.78 \times 57.3 - 102}{8.78} = 52.3$$

(5) The total coil load may be calculated from the enthalpy difference across the coil and the air quantity using the weight of dry air instead of the weight of the mixture.

$$q_t = G_a (h_1 - h_2)$$
$$= 1700\ (49.24 - 23.77) = 43{,}200 \text{ Btu per (hr) (sq ft of face area)}$$

where

G_a = Weight of dry air per (hour) (square foot of coil face area).
h_1 = enthalpy of air vapor mixture entering coil, Btu per pound of dry air.
h_2 = enthalpy of air vapor mixture leaving coil, Btu per pound of dry air.

(6) The refrigerant temperature may be found from Equation 9

$$\frac{43{,}200}{15 \times 6 \times \frac{325}{15}} = (t_s - t_r) = 22.1$$

Therefore, $t_r = (52.3 - 22.1) = 30.2$.

Thus a coil 6 rows deep operating at a refrigerant temperature of 30.2 F and a face velocity of 400 fpm is required and it will carry a total load of 43,200 Btu per (hour) (square foot of face area). The air conditions leaving the coil are too low for the conditions of the problem and therefore it is necessary to by-pass air at the entering condition to obtain the desired result of 80.5 F dry-bulb and 73 F wet-bulb.

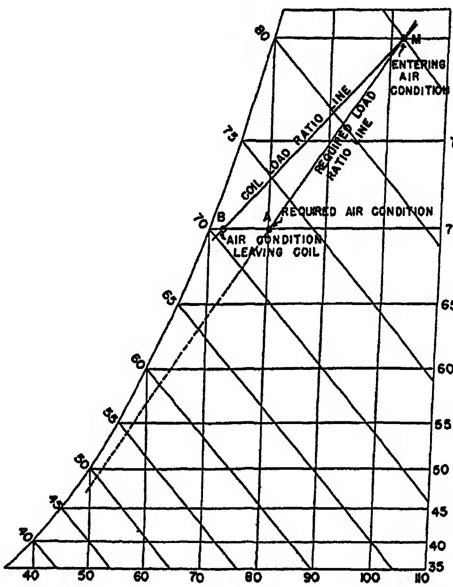

FIG. 3. PSYCHROMETRIC LAYOUT FOR COIL SELECTION

Although the preceding solution is satisfactory, it may be more desirable in some cases to use a higher refrigerant temperature and employ reheat to obtain the desired load ratio. Such a solution is shown in Fig. 3. In this case the coil load ratio line intersects the saturation curve and therefore a coil of any depth may be selected.

If a coil depth of 6 rows is maintained, the exit air conditions for the coil are indicated at point B Fig. 3 as 72.3 F dry-bulb and 70.8 F wet-bulb and the surface temperature will be:

$$t_s = \frac{9.78 \times 72.3 - 102}{8.78} = 69.0$$

The coil load will be: $q_t = 1700\ (49.24 - 34.66) = 24{,}800$ Btu per (hour) (square foot of face area) and the refrigerant temperature will be found from Equation 9:

$$\frac{24{,}800}{15 \times 6 \times \frac{325}{15}} = (t_s - t_r) = 12.7$$

Therefore, $t_r = 69.0 - 12.7 = 56.3$.

Thus, for the case where reheat is used, a coil 6 rows deep operating at a refrigerant temperature of 56.3 F is required. The total coil load will be 24,800 Btu per (hour) (square foot of face area) but the actual effective load will be less by the amount of reheat required. Therefore, for a given load, a larger coil and more refrigerating capacity are required when reheat is used.

LETTER SYMBOLS USED IN CHAPTER 7

η = fin efficiency.

A = external area of coil, square feet per (square foot of coil face area) (row of coil depth).

$B = \dfrac{t_1 - t_{dp1}}{t_2 - t_{dp2}}$

D = internal diameter of tube, inches.

G = air mass velocity, pounds per (hour) (square foot of coil face area).

G_a = dry air mass velocity, pounds dry air per (hour) (square foot of coil face area).

h_1 = enthalpy of air-vapor mixture entering coil, Btu per pound of dry air.

h_2 = enthalpy of air-vapor mixture leaving coil, Btu per pound of dry air.

h_a = film coefficient of heat transfer between air and external coil surface, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and coil).

h_r = film coefficient of heat transfer between fluid and internal coil surface, Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature between fluid and surface).

k = conductivity of pipe or tube material, Btu (square foot) (hour) (Fahrenheit degree per inch thickness).

L = thickness of tube wall, inches.

MTD = abbreviation—mean temperature difference between fluid in coil and air passing over coil, Fahrenheit degrees.
NOTE: *MTD*—usually logarithmic mean.

MTD_a = logarithmic mean temperature difference between air and coil surface.

N = number of rows of coil depth.

n = a constant, exponent of G in Equation 8, obtained by plotting, on logarithmic coordinates, G against values of h_a. The value of n is the slope of the line.

q_s = sensible heat transferred, Btu per (hour) (square foot of coil face area).

q_t = total heat transferred by coil, Btu per (hour) (square foot of face area).

R = ratio between external and internal surface of tube.

t = average water temperature, Fahrenheit degrees.

t_1 = dry-bulb temperature of air entering coil, Fahrenheit degrees.

t_2 = dry-bulb temperature of air leaving coil, Fahrenheit degrees.

t_a = minimum dry-bulb temperature possible without dehumidification, Fahrenheit degrees.

t_{dp1} = dew-point of air entering coil, Fahrenheit degrees.

t_{dp2} = dew-point of air leaving coil, Fahrenheit degrees.

t_r = average refrigerant temperature, Fahrenheit degrees.

t_s = average temperature of external surface of coil, Fahrenheit degrees.

$\Delta t = t_s - t_r$.

U = over-all coefficient of heat transfer, Btu per (hour) (square foot of external coil surface) (Fahrenheit degrees temperature difference between fluid in coil and air flowing over coil).

V = water velocity, feet per second.

Z = a constant for use in Equation 8 obtained by plotting on logarithmic coordinates G against values of h_a.

NOTE: Numerical subscripts refer to condition entering and leaving respectively.

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[3]—The Effect of Turbulence Promoters on Heat Transfer Coefficients for Water Flowing in Horizontal Tubes, by L. G. Seigel (A. S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1946, p. 111).

[4]—Maximum Rate of Heat Transfer with Minimum Loss of Energy, by Z. Nagoaka and A. Watanabe (Proceedings, *International Congress on Refrigeration*, 7th Congress, Vol. 3, No. 16, pp 221-245, 1937).

CHAPTER 8

Air Leakage

Causes of Infiltration, Infiltration Due to Wind Pressure, Infiltration Through Walls, Window and Door Leakage, Crack Method, Air Change Method, Infiltration Due to Temperature Difference, Sealing of Vertical Openings

THE air leakage which takes place through various apertures in buildings must be considered in heating and cooling calculations, and properly evaluated. This *infiltration* as it is sometimes designated takes place through cracks around doors and windows, through solid walls and through fireplaces and chimneys. Although the latter sources of leakage may be considerable, they are often neglected on the assumption that dampers would be closed during periods of extreme cold weather or else that the fireplace will be in use at such times and will therefore contribute to the heat supplied and lessen the heating load.

CAUSES OF INFILTRATION

The displacement of heated air in buildings by unheated outside air is due to two causes, namely, (1) the pressure exerted by the wind and (2) the difference in density of outside and inside air because of differences in temperature. The former is generally referred to as *infiltration* and the latter as *stack* or *chimney effect.*

In either case an exact estimate of the amount of infiltration under design conditions is difficult to make. The complicating factors include (1) variations in building construction particularly as to width of crack or size of openings through which air leakage takes place, (2) the variations in wind velocity and direction, (3) the exposure of the building with respect to air leakage openings and with respect to adjoining buildings, (4) the variations in outside temperatures which influence the chimney effect, (5) the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors, and (6) the influence of a planned air supply and the related outlet vents. Tight construction is essential for preventing large heat loss due to infiltration.

INFILTRATION DUE TO WIND PRESSURE

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages owing to this building up of pressure will be 80 per cent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Mechanical ventilating systems are frequently designed to produce positive or negative pressures in an enclosure which are greater or lower than prevalent wind pressures. In such designs, if the rate at which air is specified to be introduced to or removed from the enclosure by positive means exceeds the infiltration rate, it is common practice to use the greater value in determining the heating capacity to warm the outside air.

Infiltration Through Walls

Data on infiltration through brick and frame walls are given in Table 1[1]. The brick walls listed in this table are walls which show poor workmanship and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 per cent or a practically negligible quantity, which indicates the importance of good workmanship in proper sealing at the baseboard. It will be noted from Table 1, that the infiltration through properly plastered walls can be neglected.

The value of building paper when applied between sheathing and shingles is indicated by Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them. The infiltration indicated in Fig. 1 is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

Table 1. Infiltration Through Walls[a]

Expressed in cubic feet per square foot per hour

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
8½ in. Brick Wall	Plain	2	4	8	12	19	23
	Plastered[b]	0.02	0.04	0.07	0.11	0.16	0.24
13 in. Brick Wall	Plain	1	4	7	12	16	21
	Plastered[b]	0.01	0.01	0.03	0 04	0.07	0.10
	Plastered[c]	0.03	0 10	0.21	0 36	0.53	0.72
Frame Wall, with lath and plaster[d]		0.03	0.07	0.13	0.18	0.23	0.26

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in the papers listed in chapter footnotes.

[b]Two coats prepared gypsum plaster on brick.

[c]Furring, lath, and two coats prepared gypsum plaster on brick.

[d]Wall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and three coats gypsum plaster.

Window and Door Leakage

There are two methods of estimating air leakage through window and door cracks, namely, (1) the crack method and (2) the air change method. The crack method is generally regarded as being more accurate than the purely arbitrary air change method, provided the variables entering into the crack method, such as crack width and clearance, can be properly evaluated.

Crack Method

The crack method is based on known air leakage factors for various types of windows and widths of crack and clearance. The wind velocity and length of crack are also considered when the crack method is employed. The amount of infiltration for various types of windows is given in Table 2[2]. The fit of double-hung wood windows is determined by crack and clearance. Crack thickness is equivalent to one-half the difference between the inside window frame dimension and the outside sash width. The difference between the width of the window frame guide and the sash thickness is considered as the clearance. The length of the perimeter opening or crack for a double-hung window is equal to three times the width plus two times the height, or in other words, it is the outer sash perimeter length plus the meeting rail length. Not all the window crack in any given room is necessarily used in estimating the infiltration heat

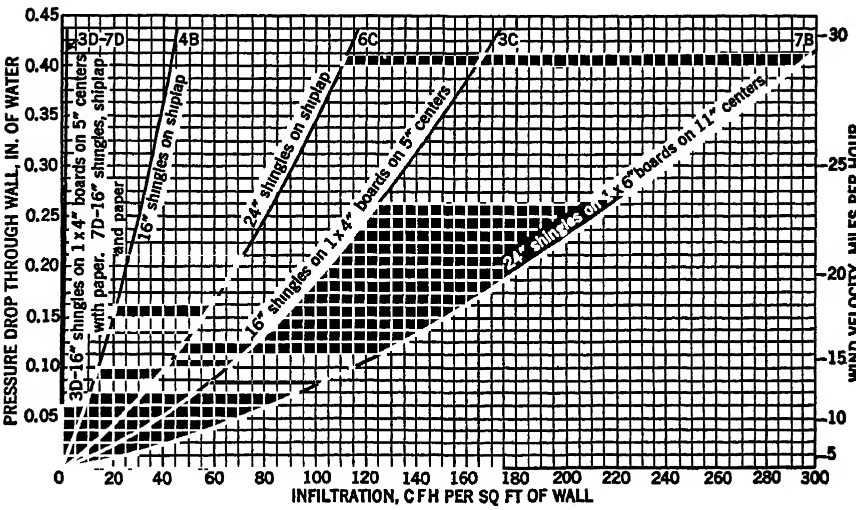

FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

loss by the crack method. The length of crack to be selected in any given case depends on the number of exposed sides as explained in Chapter 14.

Values of leakage shown in Table 2 for the average double-hung wood window were determined by using, on nine windows tested in the laboratory, the average measured crack and clearance of a large number of windows found in a field survey. In addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window

Table 2. Infiltration Through Windows

Expressed in Cubic Feet per Foot of Crack per Hour[a]

Type of Window	Remarks	Wind Velocity, Miles per Hour 5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3	8	14	20	27	35
	Around frame in masonry wall—calked[b]	1	2	3	4	5	6
	Around frame in wood frame construction[b]	2	6	11	17	23	30
	Total for average window, non-weatherstripped, 1/16-in. crack and 3/64-in. clearance.[c] Includes wood frame leakage[d]	7	21	39	59	80	104
	Ditto, weatherstripped[d]	4	13	24	36	49	63
	Total for poorly fitted window, non-weatherstripped, 3/32-in. crack and 3/32-in. clearance.[e] Includes wood frame leakage[d]	27	69	111	154	199	249
	Ditto, weatherstripped[d]	6	19	34	51	71	92
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted, 1/16-in. crack[g]	52	108	176	244	304	372
	Architectural projected, 1/32-in. crack[h]	15	36	62	86	112	139
	Architectural projected, 3/64-in. crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in. crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in. crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in. crack[j]	3	10	18	26	36	48
	Heavy casement section, projected 1/32-in. crack[j]	8	24	38	54	72	92
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table, with the exception of those for double-hung and hollow metal windows, are 20 per cent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed in chapter footnotes.

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and non-calked tests.

[c]The fit of the average double-hung wood window was determined as 1/16-in. crack and 3/64-in. clearance by measurements on approximately 600 windows under heating season conditions.

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage. The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking.

[e]A 3/32-in. crack and clearance represent a poorly fitted window, much poorer than average.

[f]Windows tested in place in building.

[g]Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

[h]Architecturally projected made of same sections as industrial pivoted except that outside framing member is heavier, and it has refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms. 1/32-in. crack is obtainable in the best practice of manufacture and installation, 3/64-in. crack considered to represent average practice.

[i]Of same design and section shapes as so-called *heavy section casement* but of lighter weight. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[j]Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With 3/64-in. crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table.

generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of sash section in contact with steel work at mullions should be figured at 25 per cent of the values for industrial pivoted windows as given in Table 2.

When storm sash are applied to well fitted windows, very little reduction in infiltration is secured, but the application of the sash does give an air space which reduces the heat transmission and helps prevent the frosting of the windows[3]. By applying storm sash to poorly fitted windows, a reduction in leakage of 50 per cent may be obtained, the effect so far as air leakage is concerned being roughly equivalent to that obtained by the installation of weatherstrips.

Door Leakage

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should be used. If weatherstripped, the values may be reduced one-half. A single door which is frequently opened, such as might be found in a store, should have a value applied which is three times that for a well fitted door. This extra allowance is for opening and closing losses and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

The infiltration rate through swinging and revolving doors is generally a matter of judgment by the engineer making cooling load determinations and in the absence of adequate research data the values given in Table 3 represent current engineering practice[4]. These values are based on the average number of persons in a room at a specified time, which may also

TABLE 3. INFILTRATION THROUGH OUTSIDE DOORS FOR COOLING LOADS[a]

Expressed in Cubic Feet per Minute per Person Entering Room

APPLICATION	PAIR 36 IN. SWINGING DOORS, SINGLE ENTRANCE[b]	APPLICATION	PAIR 36 IN. SWINGING DOORS, SINGLE ENTRANCE[b]
Bank	7.5	Hospital Room	3.5
Barber Shop	4.5	Lunch Room	5.0
Broker's Office	7.0	Men's Shop	3.5
Candy and Soda	6.0	Office	3.0
Department Store	8.0	Office Building	2.0
Dress Shop	2.5	Public Building	2.5
Drug Store	7.0	Restaurant	2.5
Furrier	2.5	Shoe Store	3.5

[a]For doors located in only one wall or where doors in other walls are of revolving type.

[b]Vestibules with double pair swinging doors, infiltration may be assumed 75 per cent of swinging door values.

Infiltration for 72 in. revolving doors may be assumed 60 per cent of swinging door values.

be the same occupancy assumed for determining the outside ventilation requirements outlined in Chapters 12 and 15.

Air Change Method

The amount of air leakage is sometimes roughly estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room, as indicated in Table 4. This method may be used to advantage as a check on the calculations made in the more exact manner. On the other hand, where it is not possible to determine or pre-determine with accuracy the width of crack or clearance of windows, or where other sources of air leakage cannot readily be evaluated, as is often the case, the use of the air change method may be justified.

The values in Table 4 may be used with reasonable accuracy for residences and are the requirements for each room. The total infiltration allowance for the entire building should be one-half the sum of the infiltration allowances of the individual rooms, since whatever air enters on the windward side generally leaves the building on the leeward side and the infiltration requirements therefore do not exist simultaneously on all sides or in all rooms. An allowance of one air change per hour for all sources of air leakage for the entire volume may be considered average for a well constructed residence.

The air leakage for vestibules due to opening and closing of doors is sometimes based on the air change method, even though the air leakage estimates for other rooms are based on the crack method. Except for vestibules and reception halls, it is not advisable to attempt to apply the air change method to factories and industrial and commercial buildings because of wide variations in the type and percentage of fenestration which is the principal source of air leakage in such buildings.

INFILTRATION DUE TO TEMPERATURE DIFFERENCE

The air exchange due to temperature difference, inside to outside, is a chimney effect, causing air to enter through openings at lower levels and to leave at higher levels[5]. Although it is not appreciable in low buildings, this loss should be considered in tall, single story buildings with openings near the ground level and near the ceiling. Also in tall, multi-story buildings it may be a considerable item unless the sealing between various floors and rooms is quite perfect.

In tall buildings, temperature difference or chimney effect will produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels. On the other hand, the wind velocity at lower

TABLE 4. AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR PROVIDED FOR VENTILATION[a]

Kind of Room or Building	Number of Air Changes taking Place per Hour	Kind of Room or Building	Number of Air Changes taking Place per Hour
Rooms, 1 side exposed	1	Rooms with no windows or outside doors	½ to ¾
Rooms, 2 sides exposed	1½	Entrance Halls	2 to 3
Rooms, 3 sides exposed	2	Reception Halls	2
Rooms, 4 sides exposed	2	Bath Rooms	2
		Stores	1 to 3

[a]For rooms with weatherstripped windows or storm sash, use ½ these values, where applicable.

levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors, thereby preventing free upward movement, so that wind and temperature difference may seldom cooperate to the fullest extent. Making the rough assumption that the *neutral zone*[6] is located at mid-height of a building, and that the temperature difference is 70 F, Equations 1 and 2 may be used to determine an equivalent wind velocity to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$V_e = \sqrt{V^2 - 1.75\,a} \tag{1}$$

$$V_e = \sqrt{V^2 + 1.75\,b} \tag{2}$$

where

V_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.
V = wind velocity upon which infiltration would be determined if temperature difference were disregarded.
a = distance of windows under consideration from mid-height of building if *above* mid-height, feet.
b = distance if *below* mid-height, feet.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist.

Sealing of Vertical Openings

In tall, multi-story buildings, every effort should be made to seal off vertical openings such as stair-wells and elevator shafts from the remainder of the building. Stair-wells should be equipped with self-closing doors, and, in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead, the greater wind movement at the greater heights makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence automatic temperature control is especially desirable with such installations.

In stair-wells that are open through many floor levels although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels and to decrease the amount at higher levels. One rule is to calculate the heating surface of the entire stair-well in the usual way and to place 50 per cent of this in the bottom third, the normal amount in the middle third and the balance in the top third.

Infiltration and Air for Combustion

Infiltration in residences normally supplies the air required for combustion by fuel burning appliances, but in some residences weather-

stripping, sealing and caulking may reduce infiltration to the point that special openings must be provided to supply adequate air to the heating appliances.

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CHAPTER 9

Natural Ventilation

Wind Forces, Temperature Difference Forces, Heat Removal, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces finds application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for moving air into, through, and out of buildings are: (*a*) wind forces, and (*b*) the difference in temperature between the air inside and outside a building. The air movement may be caused by either of these forces acting alone or by a combination of the two, depending upon atmospheric conditions, building design and location. The ventilating results obtained will vary, from time to time, due to variation in the velocity and direction of the wind and the temperature difference. The arrangement, location, and control of the ventilating openings should be such that the two forces act cooperatively rather than in opposition.

WIND FORCES

In considering the use of natural wind forces for producing ventilation, account must be taken of: (1) average wind velocity, (2) prevailing wind direction, (3) seasonal and daily variations in velocity and direction, and (4) local wind interference by nearby buildings, hills or other obstructions of similar nature.

Values are given in Table 2, Chapter 15 for the average wind velocities for the months June to September in various localities throughout the United States, while Table 1, Chapter 14, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While the tables give no average velocities below 5 mph, there will be times when the velocity is lower, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the velocity falls below one-half of the average for many hours per month. Consequently, if the natural ventilating system is designed for wind velocities of one-half of the average seasonal velocity, it should prove satisfactory in almost every case.

Equation 1 may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings to produce given results:

$$Q = EAV \tag{1}$$

where

Q = air flow, cubic feet per minute.
A = free area of inlet openings, square feet.
V = wind velocity, feet per minute, = miles per hour $\times$ 88.
E = effectiveness of openings. (E should be taken at 0.50 to 0.60 for perpendicular winds and 0.25 to 0.35 for diagonal winds[1].)

The accuracy of the results obtained by the use of Equation 1 depends upon the placing of the openings, as the formula assumes that ventilating

openings have a flow coefficient slightly greater than that of a square-edged orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less and, if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the five places listed:

1. On the side of the building directly opposite the direction of the prevailing wind.
2. On the roof in the low pressure area caused by the jump of the wind (see Fig. 1).

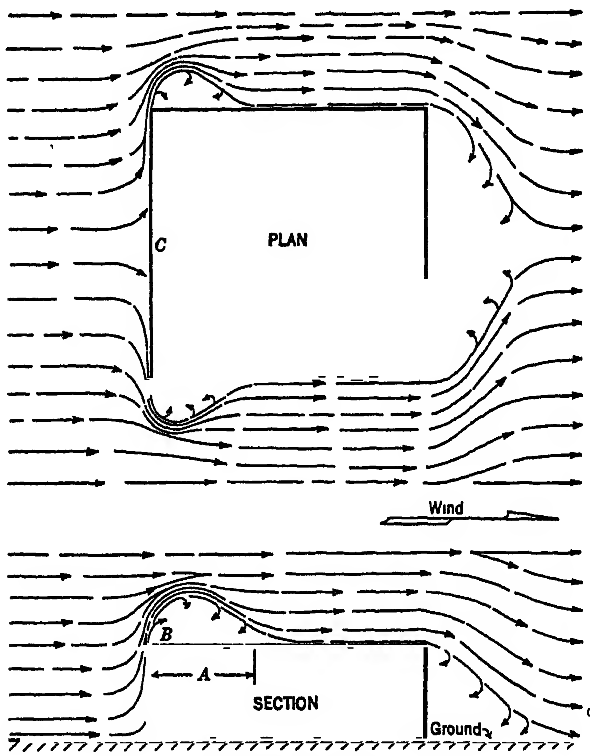

FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (*A*—LENGTH OF SUCTION AREA; *B*—POINT OF MAXIMUM INTENSITY OF SUCTION; *C*—POINT OF MAXIMUM PRESSURE)

3. On the sides adjacent to the windward face where low pressure areas occur.
4. In a monitor on the side opposite from the wind.
5. In roof ventilators or stacks.

TEMPERATURE DIFFERENCE FORCES[2]

The stack effect produced within a building when the outdoor temperature is lower than the indoor temperature is due to the difference in weight of the warm column of air within the building and cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4\,A\sqrt{h\,(t - t_0)} \qquad (2)$$

where

Q = air flow, cubic feet per minute.
A = free area of inlets or outlets (assumed equal), square feet.
h = height from inlets to outlets, feet.
t = *average* temperature of indoor air in height h, Fahrenheit degrees.
t_0 = temperature of outdoor air, Fahrenheit degrees.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings. This should be reduced to 50 per cent (constant = 7.2) if conditions are not favorable.

HEAT REMOVAL

In problems of heat removal, knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount

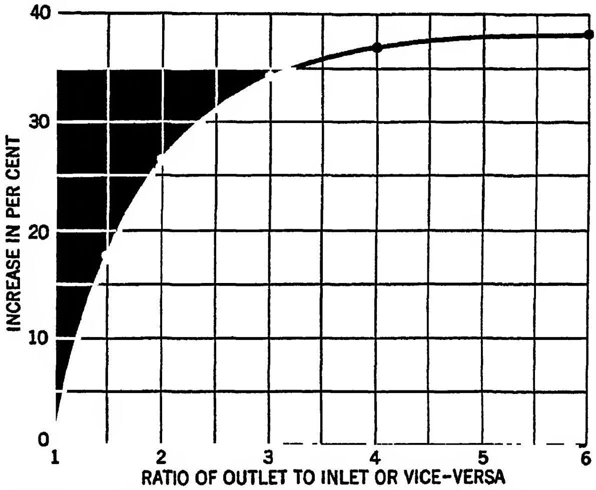

FIG. 2. INCREASE IN FLOW CAUSED BY EXCESS OF ONE OPENING OVER ANOTHER

of air to be passed through the building per minute to maintain this temperature difference can be determined by means of Equation 3.

$$H = 0.0175\, Q\, (t - t_0) \tag{3}$$

where

H = heat removed, Btu per minute.
Q = air flow, cubic feet per minute.
$t - t_0$ = inside-outside temperature difference, Fahrenheit degrees.

EFFECT OF UNEQUAL OPENINGS

The largest flow per unit area of openings is obtained when inlets and outlets are equal, and the equations given previously are based on this condition. Increasing outlets over inlets, or vice-versa, will increase the air flow, but not in proportion to the added area. When solving problems having an unequal distribution of openings, use the smaller area, either inlet or outlet, in the equations and add the increase as determined from Fig. 2.

COMBINED FORCES OF WIND AND TEMPERATURE

Equations for determining the air flow due to temperature difference and wind have already been given. It must be remembered that when

both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The flow through any opening is proportional to the square root of the sum of the heads acting on that opening.

When the two heads are about equal in value and the ventilating openings are operated so as to coordinate them, the total air flow through the building is about 10 per cent greater than that produced by either head acting independently under conditions ideal to it. This percentage decreases rapidly as one head increases over the other and the larger will predominate.

The wind velocity and direction, the outdoor temperature, or the indoor distribution, cannot be predicted with certainty, and refinement in calculations is not justified; consequently, a simplified method can be

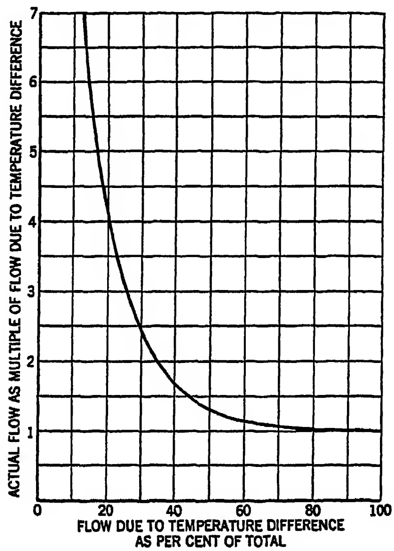

Fig. 3. Determination of Flow Caused by Combined Forces of Wind and Temperature Difference

used. This may be done by using the equations and calculating the flows produced by each force separately under conditions of openings best suited for coordination of the forces. Then, by determining, as a percentage, the ratio of the flow produced by temperature difference to the sum of the two flows, the actual flow due to the combined forces can be approximated from Fig. 3.

Example 1. Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per pound is used in this shop at the rate of 15 gal per hour (7.75 lb per gal). Desired summer temperature difference is 10 deg and the prevailing wind is 8 mph perpendicular to the long dimension. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution for Temperature Difference Only. The heat $H = \dfrac{15 \times 7.75 \times 18{,}000}{60} =$ 34,875 Btu per minute.

By Equation 3, the air flow required to remove this heat with an average temperature difference of 10 deg is:

$$Q = \frac{H}{0.0175\,(t - t_o)} = \frac{34{,}875}{0.0175 \times 10} = 199{,}286 \text{ cfm.}$$

This is equal to about 20 air changes per hour. From Equation 2 the inlet (or outlet) opening area should be:

$$A = \frac{Q}{9.4\sqrt{h\,(t - t_o)}} = \frac{199{,}286}{9.4\sqrt{30 \times 10}} = 1224 \text{ sq ft.}$$

The flow per square foot of inlet or outlet would be 199,286 ÷ 1224 = 163 cfm with all windows open.

Solution for Wind Only. With 1,224 sq ft of inlet openings distributed around the sidewalls, there will be about 410 sq ft in each long side and 202 sq ft in each end. The outlet area will be equally distributed on the two sides of the monitor, or 612 sq ft on each side. With the wind perpendicular to the long side, there will be 410 sq ft of opening in its path for inflow and 612 in the lee side of the monitor for outflow with the windward side closed. The air flow, as calculated by Equation 1, will be:

$$Q = 0.60 \times 410 \times 704 = 173{,}200 \text{ cfm.}$$

This gives 17.3 air changes per hour, which should be more than ample when there is no heat to be removed.

Solution for Combined Heads. Since the windward side of the monitor is closed when the wind is blowing, the flow due to temperature difference must be calculated for this condition, using Fig. 2. This chart shows that when inlets are twice the size of the outlets, in this case 1,224 sq ft in the sidewalls and 612 sq ft in the monitor, the flow will be increased 26.5 per cent over that produced by equal openings. Using the smaller opening and the flow per square foot obtained previously, the calculated amount for this condition will be:

$$612 \times 163 \times 1.265 = 126{,}200 \text{ cfm.}$$

Adding the two computed flows:

Temperature Difference	= 126,200	=	42 per cent.
Wind	= 173,200	=	58 per cent.
Total	299,400	=	100 per cent.

From Fig. 3, it is determined that when the flow, due to temperature difference, is 42 per cent of the total, the actual flow, due to the combined forces, will be about 1.6 times that calculated for temperature difference alone, or 201,920 cfm.

The original flow, due to temperature difference alone, was 199,286 cfm with all openings in use. The effect of the wind is to increase this to 201,920 cfm even though half of the outlets are closed.

A factor of judgment is necessary in the location of the openings in a building, especially those in the roof, where heat, smoke and fumes are to be removed. Usually windward monitor openings should be closed, but if the wind is low enough for the temperature head to overcome it, all windows may be opened.

TYPES OF OPENINGS

Types of openings may be classified as: (1) windows, doors, monitor openings and skylights, (2) roof ventilators, (3) stacks connecting to registers, and (4) specially designed inlet or outlet openings.

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area when open. Their movable parts are arranged to open in various ways; they may open by sliding either vertically or horizontally, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top, bottom or side. Regardless of their design, the air flow per square foot of opening may be considered to be the same under the same conditions. The type of pivoting should receive consideration

from the standpoint of weather protection, and certain types may be advantageous in controlling the distribution of incoming air. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet. These are actuated by the same forces of wind and temperature head, which create flow through other types of openings. The capacity of a ventilator depends upon four things: (1) its location on the roof, (2) the resistance it and the duct work offers to air flow, (3) the height of draft, and (4) the efficiency of the ventilator in utilizing the kinetic energy of the wind for inducing flow by centrifugal or ejector action.

For maximum flow induction, a ventilator should be located on that part of the roof where it will receive the full wind without interference. If ventilators are installed within the suction region created by the wind passing over the building, or in a light court, or on a low building between two high buildings, their performance will be seriously influenced. Their normal ejector action, if any, may be completely lost.

The base of the ventilator should be of a taper-cone design to produce the effect of a bell-mouth nozzle whose coefficient of flow is considerably higher than that of a square-entrance orifice. If a grille is provided at the base, or if the base or structural members present obstructions, additional resistance is introduced, and the base opening should be increased in size accordingly.

Air inlet openings located at lower levels in the building should be at least equal to, and preferably larger than the combined throat areas of all roof ventilators. The air discharged by a roof ventilator depends on wind velocity and temperature difference, and, in general, its performance will be the same as any monitor opening located in the same place but, due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Roof ventilators may be classified as stationary, pivoting or oscillating, and rotating. Generally, these have a round throat, but the continuous-ridge ventilator, or so-called heat valve, would fall in the stationary classification. When selecting roof ventilators, some attention should be given to ruggedness of construction, storm proofing features, dampers and damper operating mechanisms, possibility of noise, original cost, and maintenance.

Natural ventilation units may be used to supplement power-driven supply fans, and under favorable weather conditions it may be possible to stop the power-driven units. Units are not subject to code tests for ratings. Generally they must be selected from manufacturers' tables. It is, therefore, very important to consider the reliability of the ratings used.

Controls

Gravity ventilators may have dampers controlled by hand, thermostat, or wind velocity, in combination with a fan. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks or vertical flues are really chimneys which function through the effects of the wind and temperature difference. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction. With little or no wind, the chimney effect depends entirely on temperature difference to produce a removal of air from the rooms where the inlet openings are located.

GENERAL RULES

A few of the important requirements in addition to those already outlined are:

1. Inlet openings in the building should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top. Outside air will then be supplied to the zone to be ventilated.

2. Inlet openings should not be obstructed by buildings, trees, sign boards, etc. outside nor by partitions inside.

3. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

4. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building together with amount and grouping of ventilation openings can be readily arranged to take full advantage of the force of the wind. Where the wind's direction is quite variable, the openings should be arranged in sidewalls and monitors so that, as far as possible, there will be approximately equal areas on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force and others to a suction force, and effective movement through the building will be assured.

5. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

6. In order that temperature difference may produce a motive force, there must be vertical distance between openings. That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great that difference might be.

7. In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are less effective for ventilation.

8. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in cooperation of wind and temperature difference.

9. In an industrial building where furnaces that give off heat and fumes are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas laden air.

10. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that the wind, in splashing therefrom, will create a suction. The additional height also increases the effect of temperature difference to cooperate with the wind.

11. The intensity of suction or the vacuum produced by the jump of the wind is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when changes in occupancy are possible, or to provide for extremely hot days. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. In single story industrial buildings, particularly those covering large areas, natural ventilation must be accomplished by taking air in and out of the roof openings. Openings in the pressure zones can be used for inflow and openings in the suction zone, or openings in zones of less pressure, can be used for outflow. The ventilation is accomplished by the manipulation of openings to get air flow through the zones to be ventilated.

DAIRY BARN VENTILATION[8]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture, and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hour. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hour. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume and construction permit adequate heating by the stabled animals, the air supply need not be heated. The air should be supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hour, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cu ft per hour of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per (hour) (cubic foot of barn space), and 0.197 to 0.305 Btu per (hour) (square foot of barn exposure).

GARAGE VENTILATION

On account of the hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be overemphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means, particularly during the mild weather when doors and windows can be kept open. However, the A.S.H.V.E. Code of Minimum Requirements for Heating and Ventilating Garages, adopted in 1935, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impracticable to operate such a system of natural ventilation, a mechanical system shall be used which shall

provide for either the supply of 1 cu ft of air per minute from out-of-doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage[4].

Research

Research on garage ventilation, undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the University of Kansas, Lawrence, Kans., in cooperation with the A.S.H.V.E. Research Laboratory, and at the A.S.H.V.E. Research Laboratory, has resulted in authoritative papers on the subject.

Some of the conclusions from work at the Laboratory are listed in the following statements:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level.

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. In the average case upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage than that obtained with mixing of the exhaust gases and the air supplied. However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.

4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cu ft per hour, with an average rate of 35 cu ft per hour.

5. An air change of 350,000 cu ft per hour per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

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CHAPTER 10

Air Contaminants

Classification of Air Contaminants; Sizes of Airborne Particles; Air Pollution by Smoke, Ash and Cinders, Smoke Abatement; Odor Nuisance, Maximum Allowable Concentrations of Industrial Air Contaminants; Flammable Gases and Vapors; Combustible Dusts; Atmospheric Pollen; Airborne Bacteria

THE normal constituents of the earth's atmosphere are oxygen, nitrogen, carbon dioxide, water vapor, argon, small or negligible amounts of other inert gases, hydrogen, variable traces of ozone, and small quantities of microscopic and submicroscopic solid matter, sometimes called *permanent atmospheric impurities.* From the viewpoint of the air conditioning engineer, all other airborne substances may be termed contaminants. This term is applied preferably, however, to undesirable or chance impurities, since the occasion may arise for adding to the air controlled amounts of substances such as: solid or gaseous diluents for the prevention of explosions; germicidal mists or aerosols for bacteria control; masking substances for odor control; or a substitute for one of the normal gases, as is the case when helium is used to replace nitrogen in atmospheres for compressed air workers or divers.

The control of air quality is one of the functions of complete air conditioning, and some knowledge of the composition, concentration and properties of air contaminants under various circumstances is therefore essential.

Air contaminants arise from the normal processes of wear, erosion, windstorm, sea-spray evaporation, thermal disintegration, earthquake, volcanic eruption, combustion, manufacturing, transportation, agriculture, and the biochemical or biological processes of life. They are classified at various times as organic and inorganic, visible or invisible, microscopic or macroscopic, particulate or gaseous, toxic or harmless, beneficial or destructive. The following classification is based chiefly upon the origin or method of formation of air contaminants, using distinctions that are necessarily arbitrary in some cases.

CLASSIFICATION OF AIR CONTAMINANTS

Dusts, Fumes, and Smokes are known as solid particulate air contaminants.

Dusts are solid particles projected into the air by natural forces, such as wind, volcanic eruption or earthquake, and by mechanical or man-made processes, such as crushing, grinding, milling, drilling, demolition, shovelling, conveying, screening, bagging and sweeping. Some of these forces *produce* dust from larger masses, while the others simply *disperse* materials that are already in dust or pulverized form. Generally particles are not called *dust* unless they are smaller than about 100 microns in size. Dusts may be of *mineral* type, such as rock, ore, metal, sand; *vegetable*, such as grain, flour, wood, cotton, pollen; or *animal*, such as wool, hair, silk, feathers, leather.

Fumes are solid particles commonly formed by the condensation of vapors of solid materials and may usually be found above molten metals in industrial environments. Metallic fumes generally occur as the oxides in air because of the highly reactive nature of finely divided matter. Fumes may also be formed by sublimation, distillation, calcination, or chemical reaction, whenever such processes create airborne particles predominately below the 1 micron size. Fumes permitted to *age* tend to flocculate into clumps or aggregates of much larger size and this tendency may facilitate their removal from air under controlled conditions.

Smokes are the extremely small solid particles produced by incomplete combustion of organic substances such as tobacco, wood, coal, oil, tar and other carbonaceous materials. The term *smoke* is commonly applied to the mixture of solid, liquid and gaseous products of combustion, although the technical literature prefers to distinguish between such components as soot or carbon particles, fly-ash, cinders, tarry matter, unburned gases, and gaseous combustion products The finest particulate constituents are characteristically much less than 1 micron in size, often in the range of 0 1 to 0 3 micron.

Mists and Fogs are known as liquid particulate air contaminants.

Mists are very small airborne droplets of materials that are ordinarily liquid at normal temperatures and pressures. They may be formed by atomizing, spraying, splashing, mixing, violent chemical reaction, electrolytic evolution of gas from a liquid, or escape of a dissolved gas upon release of pressure The very small droplets expelled or atomized into the air by sneezing constitute mists containing microorganisms that become air contaminants.

Fogs are limited by some classifications to airborne droplets formed by condensation from the vapor state. This arbitrary distinction between mist and fog is of minor importance, as both terms are used to indicate the particulate state of airborne liquids (occasionally termed *aerosols*). *Fog nozzles* are so named because of their ability to produce extra fine droplets as compared to the mist from ordinary spray devices The highly volatile nature of some liquids quickly reduces their airborne droplets from the mist to the fog range, and eventually to the vapor phase until the air becomes saturated with that liquid Many droplets in fogs or clouds are microscopic and submicroscopic in size, and may be conceived as the transition state between the larger mists and the vapors.

Vapors and Gases are known as gaseous non-particulate air contaminants.

Vapors are the gaseous phase of substances that are either liquid or solid in their commonly known state, examples being gasoline, kerosene, benzene, carbon tetrachloride, mercury, iodine, camphor. Vapors may be changed to the solid or liquid form by increasing the pressure, decreasing the temperature or applying both processes simultaneously. They are removed from the air by condensation with less difficulty than are the gases.

Gases are normally formless fluids which tend to occupy a space or enclosure completely and uniformly at ordinary temperatures and pressures. The following substances, therefore, qualify as gases: oxygen, nitrogen, carbon dioxide, carbon monoxide, hydrogen, ammonia, sulfur dioxide. Gases, likewise, may be solidified or liquefied by the proper control of temperature and pressure.

The preceding classification is not suitable for the airborne living organisms, which themselves range in size from the submicroscopic viruses to the largest pollen grains, not considering the smallest insect life. Bacteria range from about 0.2 to .5 microns in size, fungus spores from 1 to 10 microns, and pollen from 5 to 150 microns.

SIZES OF AIRBORNE PARTICLES

Fig. 1 is a graphic tabulation of the properties of airborne solids and liquids arranged according to size on the micron scale. There are 25,400 microns in 1 inch.

Particles larger than 10 microns are unlikely to remain suspended in air currents of moderate strength, but settle out by gravity at speeds dependent upon the shape, the size and specific gravity of the particle, the wind velocity, the orientation of the collecting surface, and the topography. These larger particles are of major interest to the engineer in the solution of nuisance problems, but it is usually the smaller particles, or those below 10 microns, that remain in the air long enough to be of hygienic as well as economic significance.

The great bulk of industrial dust particles are of the order of 1 micron

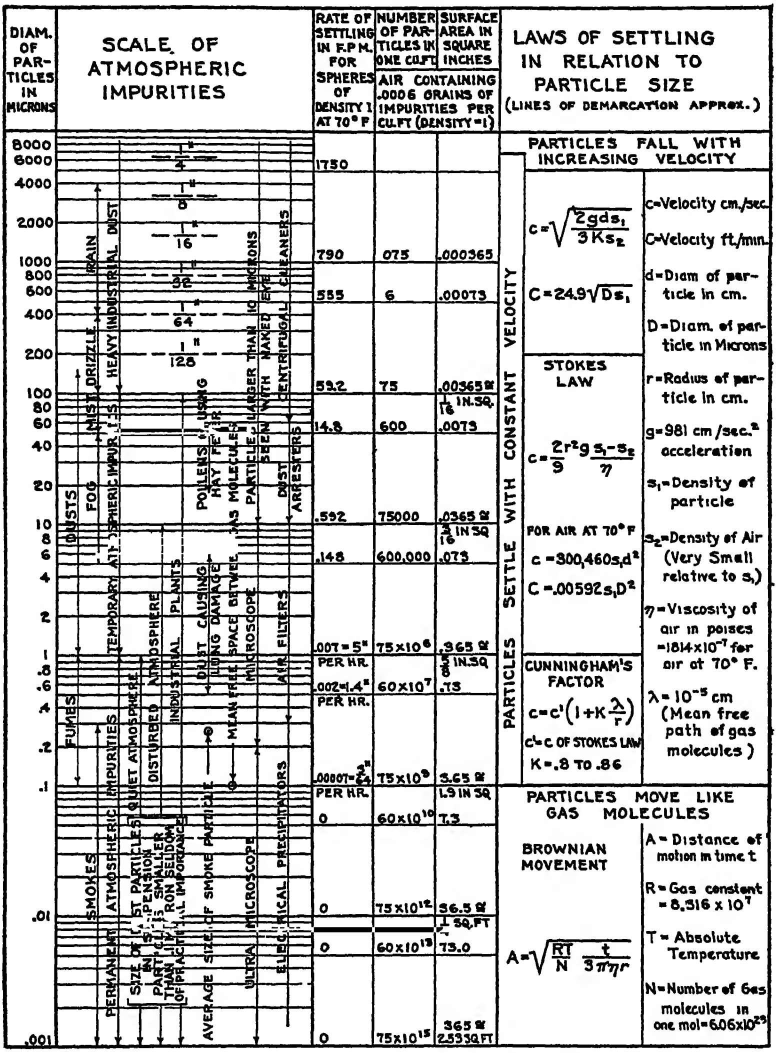

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FIG. 1. SIZES AND CHARACTERISTICS OF AIRBORNE PARTICULATE MATTER

in size. Tremendous numbers are also present in the sub-microscopic range below 0.5 micron, but those below 0.1 micron are not believed at present to be of any practical importance, possibly due to their exceedingly small mass in comparison with the balance of airborne matter. In fact, particles this small may become the *permanent* atmospheric impurities that have little if any opportunity of settling because of the continual motion imparted to them by air currents and the molecular activity of their supporting gases (Brownian Movement).

The survey[1] of atmospheric pollution in 14 American cities conducted from 1931 to 1933 indicated the average size of outdoor dust particle to be 0.5 micron, as collected by the Owens jet dust counter and measured under the microscope. The inability of the *light field* microscope to

reveal particles in the 0.1 micron vicinity may account in part for this observed statistical average of 0.5 micron.

The lower limit of particle size visible to the naked eye cannot be stated definitely. It depends not only upon the individual eye, but also upon the shape and color of the particle, the intensity and quality of the light, and the nature of the background or the opportunity for contrast. Under ideal conditions a particle of 10-micron size may be recognized, while under less favorable conditions it may be impossible to distinguish a particle smaller than 50 microns. The lower limit of visibility should, therefore, be considered as a physiological range, probably 10 to 50 microns.

Dusts, powders and granular materials are frequently classified by reference to the size of screens used for separation. Particles above 40 microns are said to be the *screen sizes* and those below, the *sub-screen* or microscopic sizes. The approximate or theoretical sizes of particles corresponding to the mesh scale of the U. S. Standard Sieve Series are given in Table 1.

TABLE 1. RELATION OF SCREEN MESH TO PARTICLE SIZE

U. S. Standard Sieve Mesh	400	325	200	140	100	60	35	18
Nominal Sieve Opening in Microns	37	44	74	105	149	250	500	1000

Microscopic examination of screened dust indicates that the average diameter of a sample of irregular particles may be substantially larger than the openings of the screen through which it has passed, if the particle shapes deviate considerably from the spherical form[2]. The smallest dimension of many such particles will correspond with the maximum permissible distance between the wires of commercial screens made to *ASTM* Standard specifications.

Screening does not give sharp separation into size groups, and accordingly such a classification is statistical rather than absolute.

Mineral particles, such as grains of sand, bits of rock, volcanic ash, or fly-ash, can be transported long distances under unusual circumstances. Thus, the dust storms of 1935 in the Kansas district resulted in vast amounts of fine top soil being thrown high into the air. Solar illumination as far east as Boston was affected noticeably and particles as large as 40 to 50 microns were actually carried half way across the continent before they settled out. In similar manner volcanic ash has been carried even farther. It is not surprising, therefore, that fly-ash from furnace gases, cement dust and the like, can be carried for considerable distances and that, occasionally, the engineer is confronted with the problem of removing such material before the air in question is suitable for use in building ventilation.

AIR POLLUTION BY SMOKE, ASH AND CINDERS

The total airborne solids settling in urban areas are usually reported as *soot fall* in tons per (square mile) (month). Such data published for the cities in this country range from 20 to 200 tons per (square mile)

(month). To the air conditioning engineer this information may indicate the effectiveness of smoke abatement or fuel combustion control methods in his locality, but it does not provide a suitable index of the suspended dust that air cleaners in a ventilating system are expected to capture [3,4,5]. Gravimetric or weight data of the type given in Table 2 are preferable. In some cases airborne particle counts may be necessary, as for pollen, bacteria, spores, and dusts causing illness or lung disease.

Dust concentrations by weight cannot be converted directly to concentrations by particle count because of the variability of particle size, shape and specific gravity, and the inherent characteristics of dust counting and weighing procedures. One milligram of dust per cubic meter of air may represent dust counts from 1 million to 100 million particles per cubic foot of air (lightfield microscope technic) according to the size distribution of the airborne dust sample. Information of this type for a specified application is best obtained by simultaneous sampling for both counting and weighing and noting carefully at the time all factors that might affect the reproducibility of the count-weight ratio.

TABLE 2. DUST CONCENTRATION RANGES

Location	Grains per 1000 Cu Ft[a]	Milligrams per Cubic Meter
Rural and suburban districts	0.02–0.2	0.05–0.5
Metropolitan districts	0.04–0.4	0.1 –1.0
Industrial districts	0.1 –2.0	0.2 –5.0
Ordinary factories or workrooms	0.2 –4.0	0.5 –10
Excessively dusty factories or mines	4–400	10–1000
Minimum explosive concentrations	4000–200,000	10,000–500,000

[a]1 grain per 1000 cu ft = 2.3 milligrams per cubic meter.
1 oz per cubic foot = 1 gram per liter = 1000 grams per cubic meter.

Smoke Abatement

Successful abatement of atmospheric pollution caused by smoke requires the combined efforts of the combustion engineer, industrial executive, public health officer, city planning commission and the community at large. Electrification of industry and railroads, increase in the use of domestic oil and gas furnaces, and segregation of industrial districts is gradually providing effective aid in the solution of this problem.

In the large cities where nuisance from smoke, fly-ash and cinders is more serious, limited areas obtain some relief by the use of district heating. Boilers in these plants are of large size, designed and operated to burn the fuel without wasteful smoke, and equipped in some cases with dust collecting devices. The gases of combustion are usually discharged at a much higher level than is possible in the case of scattered buildings equipped with separate boiler plants.

Time, temperature and turbulence are fundamental requirements for smokeless combustion. Increase of any one of these factors will reduce the quantity of smoke discharged, although excessive turbulence in furnaces may increase the output of ash and cinders. Special care must be taken in hand firing the bituminous coals. (See Chapters 16, 17 and 18 for further discussion on fuel burning technic).

Legislative measures at the present time are largely concerned with reduction of the visible smoke discharged from chimneys of boiler plants. Practically all ordinances limit the number of minutes in any one hour

that smoke of a specified density may be discharged, as measured by comparison with a Ringelmann Chart (Chapter 11, Instruments and Measurements). Ordinances generally do not make specific provision for control of the corrosive and irritant gases, such as sulfur dioxide and trioxide, which are discharged with the gases of combustion. Where high sulfur coals are burned, these sulfur gases present a serious hazard to property, health and vegetation.

In foggy weather the accumulation of these gases in the lower strata of the atmosphere may be such as to cause irritation of the eyes, nose and respiratory passages, and possibly even more dangerous consequences. The Meuse Valley (Belgium) fog disaster has become a classic example in the history of gaseous air pollution. Released in a rare combination of atmospheric calm and dense fog, it is believed that sulfur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons.

Dusts and cinders in flue gas may be caught by various available devices, such as settling chambers, centrifugal separators, electric precipitators, and gas scrubbers. The difficulty of retaining the dust and cinder particles is principally a function of their size, specific gravity, and resistance to wetting if a washer is used. Descriptions of dust collecting devices are given in Chapter 33.

Absorption of Solar Radiation

The loss of light, particularly the absorption of solar ultraviolet light by smoke and soot, is recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore [6] by actinic methods demonstrated that ultraviolet light in the country was 50 per cent greater than in the city. In New York City [7] a loss as great as 50 per cent in visible light was found by the photo-electric cell method.

ODOR NUISANCE

A problem companionate with smoke abatement is the control of odor nuisance in the neighborhood of industrial plants discharging noxious or offensive air contaminants. Community planning and zoning will avoid much of the difficulty in the future, but meanwhile many industrial cities must resort to corrective measures by requiring the installation of air cleaning devices, the alteration of manufacturing processes, or by legal termination of the offensive operation in residential or commercial districts.

The development and manufacture of more effective and economical air cleaning devices for application to industrial plants will make it possible for many industries to continue operation in their original locations if desired, even though the community standards of air purity become more rigid. To a limited extent such developments already are benefiting the manufacturer, his employees, and his neighbors, when cleaning of the air from ventilated dusty processes makes it possible to return the reclaimed or recovered air to the workroom atmosphere instead of discharging it to the outside with its full load of contamination (Chapter 33).

The control of outdoor odor nuisance is especially troublesome because of the extremely minute quantities of contaminant that are capable of offending through a wide area. New industrial chemicals with strange

or unfamiliar odors tend to receive much more attention from the neighborhood than the customary odors generated by well known processes and raw materials. Methods of odor control currently in use include charcoal adsorption, scrubbing towers and air washers, chlorination, condensation, masking, passage of the odorous air through combustion chambers, and best of all, substitution of less offensive materials whenever possible [8, 9, 10, 11].

The control of air quality within buildings ventilated for human occupancy is discussed in Chapter 12. Tobacco smoke odors, cooking odors and body odors are air contaminants of the nuisance type which now command a decisive position in the standards of air quality for indoor comfort. However, the engineer will find, at times, that odors originating outside buildings in industrial or business districts may have an even greater bearing than indoor contamination on the kind and capacity of equipment he must provide for a high quality air supply installation.

INDUSTRIAL AIR CONTAMINANTS

Many industrial processes are sources of contaminants. Their undesirable effects are known to the public and their control is an important function of the ventilating or air conditioning engineer, because the atmosphere within buildings is the medium whereby such finely divided matter is dispersed and transported from the source to remote locations where it may cause property damage, nuisance, fire, explosion, disease and even death.

Tables 3, 4 and 5 give the maximum allowable concentrations for industrial air contaminants as currently accepted in most sections of the country. They apply to *exposures of 8 hours per day*, and refer to the quantities of contaminant permissible in the *workers' breathing zone*. Some of these figures may be altered as the result of continuous research, and some may differ from those in force in a few cities or states. The prudent engineer will design equipment using these values as the *upper* limits of air contamination, and will incorporate a reasonable margin of safety in his estimates of ventilation capacity.

Information on the properties and effects, with respect to health, of specific industrial air contaminants has developed rapidly within the past decade into an extensive literature. Some of the more readily available publications are listed at the end of this chapter.

FLAMMABLE GASES AND VAPORS

Adequate ventilation is a primary requirement for eliminating or minimizing the hazard of fire or explosion due to gases and vapors. The need for good ventilation is not removed by the use of other precautions, such as the elimination of known ignition sources, segregation of hazardous operations, adoption of safe building construction, and installation of automatic alarms. Some safety engineers regard overventilation of an operation employing flammable liquids as a legitimate operating charge for the privilege or necessity of using a dangerous process. However, it is not possible to apply a reasonable safety factor to the ventilation estimate without consideration of the concentrations of gases or vapors that approach the danger point. Safety engineers prefer to limit the concentration to ¼ or ⅕ of the lower explosive limit, and this fact should be given full weight in determining the capacity and design of ventilating equipment. Rarely should consideration be given to *operation above*

TABLE 3. PHYSIOLOGICAL RESPONSE TO GASES AND VAPORS[a]
Concentrations in Parts of Substance per Million Parts of Air (ppm)

SUBSTANCE	RAPIDLY FATAL	DANGEROUS TO LIFE IN ½ TO 1 HR	MAXIMUM ALLOWABLE CONCENTRATION FOR DAILY EXPOSURES
Acrolein	2,000	100	1
Ammonia	5,000	2,500	100
Amyl acetate	----	----	400
Aniline	----	200	5
Arsine	250	10	1
Benzene (benzol)	20,000	5,000	100[b]
Bromine	500	40	1
Butyl acetate	----	10,000	400
Carbon dioxide	100,000	50,000	5,000
Carbon disulfide	2,000	1,000	20[b]
Carbon monoxide	4,000	1,000	100[b]
Carbon tetrachloride	50,000	----	100[c]
Chlorine	1,000	10	1
Dichlorobenzene	----	----	75
Dichloroethyl ether	----	500	15
Ether (diethyl)	----	35,000	400
Ethyl acetate	40,000	10,000	400
Ethylene dichloride	----	4,000	100
Formaldehyde	----	----	10[b]
Gasoline	----	----	1,000
Hydrogen chloride	1,000	----	10
Hydrogen cyanide	200	100	20
Hydrogen fluoride	----	50	3
Hydrogen sulfide	600	200	20[b]
Methyl alcohol	----	----	200[b]
Methyl bromide	20,000	2,000	50
Methyl chloride	150,000	20,000	100
Methylene chloride	----	----	500
Monochlorobenzene	----	----	75
Nitrobenzene	----	----	5
Nitrogen oxides	300	100	25[b]
Phosgene	50	5	1
Phosphine	1,000	400	1
Styrene	----	----	400[b]
Sulfur dioxide	400	150	10
Tetrachloroethane	7,000	----	10
Tetrachloroethylene	----	----	200
Toluene (toluol)	20,000	5,000	200
Trichloroethylene	----	----	200
Turpentine	----	----	200
Xylene (xylol)	20,000	5,000	200[b]

[a]Adapted from: Manual of Industrial Hygiene, by W. M. Gafafer et al, U S. Public Health Service (W. B. Saunders Co., 1943); Analytical Chemistry of Industrial Poisons, Hazards and Solvents, by M. B. Jacobs (Interscience Publishers, 1941); Noxious Gases, by Henderson and Haggard (Reinhold Publishing Co., N. Y., 1943); and other authoritative sources.

[b]Adopted by the *American Standards Association* (American Standard Z-37).

[c]50 ppm recommended by the Detroit Bureau of Industrial Hygiene.

the upper explosive limit in the open areas of buildings or rooms—even though unoccupied—because the danger of temporary drop of gas concentration to a point within the explosive range is too great.

The ability of a flammable liquid to form explosive mixtures is determined largely by its vapor pressure, volatility, or rate of evaporation.

TABLE 4. MAXIMUM ALLOWABLE CONCENTRATIONS OF DUSTS, FUMES AND MISTS[a]

SUBSTANCE	MILLIGRAMS PER CUBIC METER, DAILY EXPOSURES[b]
Arsenic; arsenic trioxide	0 15[c]
Cadmium, and compounds	0 1[c]
Chlorodiphenyls	1 0
Chromic acid mist	0.1
Lead; lead carbonate; lead chloride; lead nitrate; lead oxides; lead sulfate	0.15[c]
Manganese, and compounds	6 0[c]
Mercury, and compounds	0.1
Pentachloronaphthalene	0.5
Trichloronaphthalene	5.0
Zinc oxide fume	15 0

[a]Adapted from: Manual of Industrial Hygiene, by W. M. Gafafer, et al, U. S Public Health Service (W B Saunders Company, 1943); and other authoritative sources.

[b]1 milligram per cubic meter = 0 44 grain per 1000 cu ft.

[c]Adopted by the *American Standards Association* (American Standard Z-37).

TABLE 5. MAXIMUM ALLOWABLE CONCENTRATIONS OF DUSTS[a]

SUBSTANCE	MILLION PARTICLES PER CUBIC FOOT OF AIR, DAILY EXPOSURES[b]
Aluminum oxide abrasive	15– 50
Asbestos	5
Carborundum (silicon carbide)	15– 50
Cement (Portland)	50–100
Coal (less than 5 per cent quartz)	50–100
Dusts containing less than 10 per cent *free* silica	15–100
Granite (Barre)	10– 20
Gypsum (hydrated calcium sulfate)	50–100
Limestone (calcium carbonate)	50–100
Marble (calcium carbonate)	50–100
Mica	10– 50
Nuisance dusts (non-toxic, non-silica)	50–100
Quartz (silicon dioxide)	5
Sand (silica, silicon dioxide)	5
Sandstone	5
Silica (*free* or uncombined silicon dioxide)	5
Silicates (combined silicon dioxide)	15–100
Slate	15– 50
Talc	10– 50
Total (maximum concentration for mixed dusts)	50–100

[a]Adapted from Study of Asbestosis in Asbestos Textile Industry, U S. Public Health Service Bulletin No. 241, 1938; Industrial Dust, by Drinker and Hatch (McGraw Hill Book Co , 1936); Industrial Code Bulletin No 35, New York State Department of Labor; recommendations of state and local industrial hygiene agencies compiled by the National Conference of Governmental Industrial Hygienists; and other authoritative sources

[b]Includes only particles from 1 to 10 microns approximately, as determined by the light field microscope counting technic, using the 10× objective. Dark field counts (and the corresponding allowable concentrations) are anywhere from 2 to 100 times the light field counts for the same sample, according to the proportion of dust smaller than 1 micron (See Industrial Dust, Chapter VII, by Drinker and Hatch, McGraw Hill Book Co.).

Flash point is a convenient method of expressing this property in terms of the temperature scale. It may be defined as the temperature to which a combustible liquid must be heated to produce a flash of flame when a small flame is passed across the surface of the liquid. The higher the

flash point, the more safely can the liquid be handled. Liquids with flash points under 70 F should be regarded as highly flammable.

The upper and lower limits of flammability of gases and vapors, and the flash points of the corresponding liquids are given in Table 6.

Methods for estimating the flammable limits of mixtures of gases or vapors must be applied with caution; the reader is referred to other publications for this information [12,13].

Design of equipment for the control of combustible anesthetics is outlined in Chapter 13. Construction of equipment for handling air containing flammable substances, or operating in atmospheres so contaminated, is discussed in Chapter 46.

It is customary to report the concentrations of flammable gases or vapors in per cent by volume, or *volume per cent.* Comparison with concentrations on the *part per million* scale used in chemical, medical or industrial hygiene literature is readily made by the conversion: 1 per cent = 10,000 ppm (parts of contaminant per million parts of air, or in other words, cubic feet of contaminant per million cubic feet of air). It will be noted in Table 6 that nearly all of the substances listed have lower explosive limits above 1.0 per cent, while the *maximum allowable concentrations* for gases and vapors in Table 3 are below 1000 ppm or 0.1 per cent in most cases. Therefore, control of toxic or injurious vapors in workrooms to levels below their maximum allowable concentrations for health usually requires much more effective ventilation than for the prevention of a fire hazard.

COMBUSTIBLE DUSTS

A dust explosion is essentially a sudden pressure rise caused by the very rapid burning of airborne dust. The primary explosion often originates from a small amount of dust in suspension exposed to a source of ignition and the pressure and vibration it creates may be sufficient to dislodge large accumulations of dust on horizontal ledges or surfaces of the building and equipment, thereby creating a secondary explosion of great force. Thus the air conditioning engineer is involved for two reasons: (1) to obtain a movement of dust-laden air into exhaust hoods or openings and through ventilating or pneumatic conveying ducts in a manner that will prevent accumulation of highly flammable dust at points where it could ignite inside the equipment; and (2) to so design *process ventilation* as to prevent the escape of dust which might settle on horizontal surfaces and become a potential source of disaster at some distance from the dusty operations. (See Chapter 46).

The intensity of a dust explosion depends upon: the chemical and thermal properties of the dust; the particle size and shape; the concentration in air; the proportion of inert dust in the air; the moisture content and composition of the air; the size and temperature of the ignition source; and the degree of dispersion of the dust cloud. Investigations on the explosibility of dusts require a determination of the maximum pressure developed during an explosion of a known air concentration, as well as determination of the rate of pressure rise. Investigators frequently experience difficulty in obtaining dust suspensions of uniform dispersion, and this fact must be weighed when comparing results from several sources [14].

The minimum explosive concentrations of airborne dusts already tested range from 0.01 to 0.5 oz per cubic foot, or 10 to 500 grams per cubic

Table 6. Approximate Limits of Flammability of Single Gases and Vapors In Air at Ordinary Temperatures and Pressures[a]

Gas or Vapor	Lower Limit Per cent by Volume	Upper Limit Per cent by Volume	*Closed Cup* Flash Point F Deg[c]
Acetaldehyde	4.0	57	−17
Acetone	2.1	13.0	0
Acetylene[b]	2.5	80	......
Allyl alcohol	2.4	......	70
Ammonia	16.0	27.0	......
Amyl alcohol	1.2	......	91
Amyl chloride	1.4	......	......
Amylene	1.6	......	......
Benzene (benzol)	1.4	8.0	12
Benzyl chloride	1.1	......	140
Butane	1.6	8.5	−76
Butyl acetate	1.7	15.0	72
Butyl alcohol	1.7	......	84
Butylene	1.7	9.0	......
Carbon disulfide	1.0	50	−22
Carbon monoxide	12.5	74	......
Crotonaldehyde	2.1	15 5	55
Cyclohexane	1.3	8.4	1
Cyclopropane	2.4	10.3	......
Decane	0.67	2.6	115
Dichloroethylene (1, 2)	9.7	12.8	43
Diethyl selenide	2.5	......	......
Dioxan	2.0	22.2	65
Ethane	3 1	15.0	......
Ether (diethyl)	1.7	48.0	−20
Ethyl acetate	2.2	11.5	24
Ethyl alcohol	3.3	19.0	55
Ethyl bromide	6.7	11.3	......
Ethyl cellosolve	2.6	15 7	104
Ethyl chloride	3.6	14.8	−58
Ethylene	3.0	34.0	......
Ethylene dichloride	6.2	15.9	56
Ethyl formate	2.7	16.5	−4
Ethyl nitrite	3.0	......	−31
Ethylene oxide	3.0	80	......
Furfural (125 C)	2.1	......	140
Gasoline	1.3	6.5	−50
Heptane	1.0	6.0	25
Hexane	1.2	6.9	−7
Hydrogen cyanide	5.6	40.0	0
Hydrogen	4.1	74	......
Hydrogen sulfide	4.3	45.5	......
Illuminating gas	5.3	31.0	......
Iso-butyl alcohol	1.7	......	82
Iso-pentane	1.3	......	......
Iso-propyl acetate	1.8	7.8	43
Iso-propyl alcohol	2.5	......	53
Methane[b]	5.0	15.0	......

[a]Adapted from Limits of Inflammability of Gases and Vapors, by H. F. Coward and G. W. Jones (U. S Bureau of Mines, Bulletin No. 279, 1939); Properties of Flammable Liquids, Gases and Solids (Associated Factory Mutual Fire Ins. Cos., January, 1940); and National Fire Codes for Flammable Liquids, Gases, Chemicals and Explosives—1945 (*National Fire Protection Association*).
[b]Turbulent mixture.
[c]*Closed cup* refers to the equipment used in flash point determinations.

TABLE 6. APPROXIMATE LIMITS OF FLAMMABILITY OF SINGLE GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[a] (Continued)

GAS OR VAPOR	LOWER LIMIT PER CENT BY VOLUME	UPPER LIMIT PER CENT BY VOLUME	*Closed Cup* FLASH POINT F DEG
Methyl acetate	3.1	15.5	15
Methyl alcohol	6.0	36.5	54
Methyl bromide	13.5	14.5	....
Methyl butyl ketone	1.2	8.0	....
Methyl chloride	8.0	19.7	....
Methyl cyclohexane	1.1	....	25
Methyl ethyl ether	2.0	10.1	−35
Methyl ethyl ketone	1.8	11.5	30
Methyl formate	5.0	22.7	−2
Methyl propyl ketone	1.5	8.2	....
Natural gas	4.8	13.5	....
Naptha (benzine)	1.1	6.0	20–45
Naphthalene	0.9	....	174
Nonane	0.74	2.9	88
Octane	0.84	3.2	56
Paraldehyde	1.3	....	....
Pentane	1.4	8.0	....
Propane	2.4	9.5	....
Propyl acetate	1.8	8.0	58
Propyl alcohol	2.5	....	59
Propylene	2.0	11.1	....
Propylene dichloride	3.4	14 5	59
Propylene oxide	2.1	21.5	....
Pyridine (70 C)	1.8	12.4	68
Toluene (toluol)	1.3	7.0	40
Turpentine	0.8	....	95
Vinyl ether	1.7	27.0	....
Vinyl chloride	4.0	22.0	....
Water gas (variable)	6.0	70	....
Xylene (xylol)	1.0	6.0	63

meter of air. Maximum pressures generated have been reported as high as 500 psi, although they are more likely to be of the order of 50 psi. Investigations on the flammable characteristics of dusts are currently made at 0.1 and 0.5 oz per cubic foot[15-21].

ATMOSPHERIC POLLEN

The properties of pollen grains discharged by weeds, grasses and trees and responsible for hay fever are of special interest to engineers who design equipment for their removal from indoor air (see Allergic Disorders in Chapter 13, and Air Cleaning Devices, Chapter 33). Whole grains and fragments transported by the air range chiefly between 10 and 50 microns in size, but some have been measured as small as 5 microns and others over 100 microns in diameter. Ragweed pollen grains are fairly uniform in size within the range of 15 to 25 microns.

Pollen grains can be removed from the air more readily than the particles of dust prevalent in outdoor air and found near dusty industrial processes, since the latter predominate in the range of 0.1 to 10 microns in size.

Most grains are quite hygroscopic and therefore vary in weight with the

humidity. The specific gravity of air-dried pollen is reported to vary from 0.4 to 1.2, and accordingly their weight will either increase or decrease with the addition of moisture[22]. Their shapes and surface designs are marvelously complex, though they tend to be spherical, especially when fully distended with moisture. Illustrations and data on individual pollen grains are available in the botanical literature[22, 23, 24]. The geographical distribution of plants known to produce hay fever is also recorded[25, 26].

The quantity of pollen grains in the air is generally estimated by exposing an adhesive-coated glass plate outdoors for 24 hr and then counting calibrated areas under the microscope. Methods are available for determining the number of grains in a measured volume of air[25, 27, 28] but their greater accuracy has not caused them to replace the more simple *gravity slide method* used for most pollen counts. Counting technics vary somewhat, but the daily pollen counts reported in local newspapers during the hay fever season usually represent the number of grains found on 1.8 sq cm of a 24-hr *gravity slide*.

Hay fever sufferers may notice the first symptoms when the pollen count is 10 to 25, and in some localities the maximum figures for the seasonal peak may approach 1000 for a 24-hr period, depending upon the sampling and reporting methods of the laboratory. Translation of gravity counts by special formulas to a volumetric basis, or the number of grains per cubic yard or per cubic foot of air, is still uncertain because of the complexity of the modifying factors. When such information is important, it is best obtained directly by a volumetric instrument. The number of pollen grains per cubic yard of air evidently varies from 2 to 20 times the number found on 1 sq cm of a 24-hr gravity slide, depending on grain diameter, shape, specific gravity, wind velocity, humidity and physical placement of the collecting plate[29, 30, 31].

AIRBORNE BACTERIA

Study of the occurence and significance of micro-organisms in the atmospheres of the indoor world is currently absorbing the energies of a substantial number of physicians, bacteriologists, aerobiologists, physicists, public health workers, engineers and hospital personnel. Some data are available on the types and quantities of bacteria found in a variety of occupied and unoccupied spaces, but it is not possible at present to use this information as a conclusive index of the potential health hazard of a given environment. The number of airborne organisms may vary from 1 to 1000 per cubic foot of air, depending on the method of testing[32]. Many are attached to the dust particles which also inhabit the air.

Where it seems advisable or desirable to control the bacterial content of rooms, public conveyances or buildings, highly effective methods are available (see Chapter 13), and their extended use may do much to assist the workers in this field in accumulating the necessary mass of evidence that will decide the practical value of air sterilization for the control of communicable disease. It is now well established that ultraviolet radiation is commercially feasible for the protection or preservation of pharmaceuticals, cosmetics, and food products.

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CHAPTER 11

Instruments and Measurements

Temperature Measurement, Pressure Measurement, Measurement of Air Movement, Air Change Measurements, Measurement of Relative Humidity, Dust Determination, Heat Transfer Through Building Materials, Measurement of Heat Exchange for Comfort Conditions, Combustion Analysis, Smoke Density Measurements

THIS chapter presents a description of many test instruments used for heating, ventilating and air conditioning tests and presents a discussion of their use.

TEMPERATURE MEASUREMENT

Changes in the intensity of heat may be determined by several methods [1] such as measuring the change in volume of a liquid, the change in internal pressure of a confined gas, the current set up between dissimilar metals joined in a circuit, or the change in resistance of an electrical circuit.

Thermometers

The most common method used is the change in volume of a liquid such as mercury or alcohol enclosed in glass. Mercurial thermometers may be used for measuring temperatures from −40 F to approximately 1000 F. The lower limit is set by the freezing point of mercury. Since the boiling point of mercury is only about 675 F, the space above the mercury in thermometers designed for higher temperatures must be filled with an inert gas under pressure. Alcohol thermometers may be used for temperatures from −94 F to +248 F.

The more accurate thermometers have divisions etched on the stem and are preferred for test purposes due to their low heat capacity. During their manufacture, the freezing and boiling points of water are first marked while the thermometer is immersed in a test bath. The required divisions between these points, whether Centigrade or Fahrenheit, are then marked and etched on the stem. In spite of these precautions the probable error in etched stem thermometers is plus or minus one scale division, which makes calibration after manufacture necessary for most test work. Such thermometers are usually calibrated for complete stem immersion.

When incompletely immersed, a stem correction should be made. At ordinary atmospheric temperatures the correction is negligibly small, but it usually is important when measuring high temperatures such as those of steam and flue gas. The emergent stem correction may be calculated by the equation:

$$K = 0.00009\, D\, (t_1 - t_2) \tag{1}$$

where

K = correction to be added, Fahrenheit degrees

D = number of degrees on the thermometer scale which are not immersed.

t_1 = temperature indicated on the thermometer, Fahrenheit degrees.

t_2 = temperature of the non-immersed mercury column, Fahrenheit degrees.

0.00009 = difference in the coefficient of expansion of the mercury and glass.

Since the bulb has considerable area, radiant energy may affect temperature readings[2]. In measuring room temperatures, care must be taken to locate thermometers away from hot surfaces, such as radiators, or cold surfaces, such as walls or windows. Where this is impracticable, bright, polished shields should be used to screen the bulb from the radiant energy.

Errors may also be avoided: (*a*) by allowing the thermometer time to reach equilibrium, (*b*) by providing sufficient circulation to give a true average temperature of the medium observed, and (*c*) by reading with the eye at the same level as the top of the liquid, *i.e.*, avoiding parallax.

Two types of industrial thermometers for permanent installation in ducts or pipes are shown in Fig. 1, and consist of a glass thermometer protected by a metal case. The better grades have metal scales made specially for each instrument. These are ruled in the same manner as

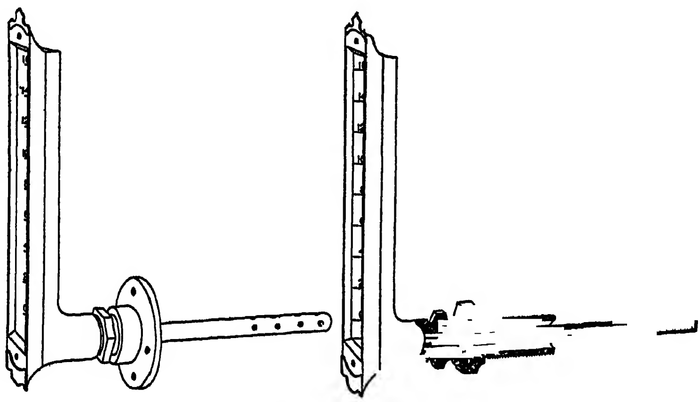

FIG. 1. INDUSTRIAL THERMOMETERS FOR PERMANENT INSTALLATION

described for etched stem thermometers. Due to the heat capacity and heat conductance of the jacket, it is more difficult to obtain the true temperature at a point with these than with the exposed etched stem type. The latter is usually preferred for test purposes.

Thermometer Wells

Where temperatures of fluids or gases in vessels or conduits are to be measured, it is often necessary to resort to thermometer wells. These are especially designed to contain thermometers and thermocouples. Since they separate the temperature sensitive element from the medium to be observed, large errors may result from their improper use[3]. These errors may be due both to poor heat transfer from the wall of the well to the sensitive element, and also to the tendency of heat to travel along the length of the well itself. To improve heat transfer in the case of thermometers, the void should be filled with a liquid of minimum practical viscosity. At the well mouth, the stem should be packed to check evaporation and heat transfer to the atmosphere. To reduce the temperature gradient over the length of the well, the wall of the conduit should be carefully insulated in the area surrounding the observation point.

Thermocouples

When two dissimilar metals are joined at two points and a temperature difference exists between these junctions, an electromotive force is developed. By maintaining the cold junction at a constant temperature (or providing equivalent electrical compensation), the magnitude of the electromotive force developed is a direct measure of the temperature of the warm junction. By proper selection of metals, any temperature up to 2900 F may be measured. Readings are obtained by means of a potentiometer or sensitive galvanometer which may be calibrated in degrees. If a potentiometer is used, there is no current flowing through the thermocouple wires at the time of measurement and therefore the resistance of the thermocouple circuit does not affect the accuracy of the reading. If a galvanometer is used instead of a potentiometer, the instrument is usually built with a high internal resistance to minimize the effect of the resistance of the thermocouple circuit and, also, special

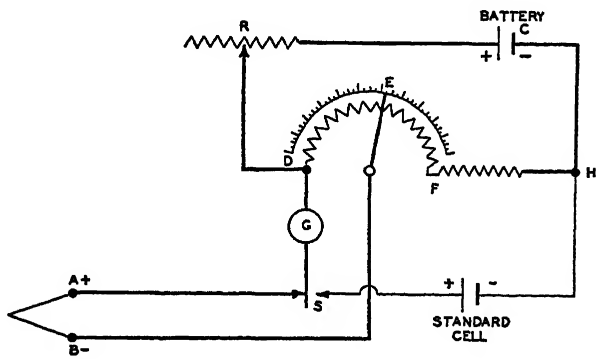

Fig. 2. Basic Circuit and Connections for Thermocouple and Potentiometer

lead wires of known resistance are used to connect the galvanometer to the thermocouples. The basic potentiometer circuits are shown in Fig. 2. The thermocouple leads A-B are so connected that their polarity opposes that of battery C. If the position of E on the graduated slide wire rheostat DF is adjusted until galvanometer G shows no current flowing, resistance DE will indicate directly the voltage generated by the thermocouple. In order to correct for variations in voltage from battery C, switch S is momentarily thrown over to the standard cell circuit and rheostat R is then adjusted so that the galvanometer shows zero current. Battery C then exerts the known voltage of the standard cell at DH.

Calibration of thermocouples for high temperatures may be made against known melting points of metals. Radiation effects may be minimized by using *the smallest size of wires consistent with mechanical strength.* The use of small wires also makes the thermocouple sensitive to minute fluctuations in temperature. Wires of No. 33 B & S gage are small enough to be sensitive and to minimize radiation effects while still rugged enough for many purposes. By the use of thermocouples, temperatures at remote points may be indicated or recorded on conveniently located instruments; average temperature may be readily obtained by connecting several couples in parallel or in series; and tem-

peratures may be obtained within thin materials, narrow spaces, or otherwise inaccessible locations.

Thermocouples in series with every alternate junction maintained at a common temperature will give an emf which, divided by the number of couples to give the average emf [4] per couple, may be used to find the average temperature.

Thermocouples in parallel having the similar metals of a number of couples connected together and run to a common cold junction will cause an indication on a potentiometer which is the true emf only if the electrical resistances of the parallel junctions are the same [4, 5].

The temperature of the surface is at best difficult to obtain accurately [6]. The thermocouple is most readily adaptable for this purpose. In a metal surface, a common method is to peen the couple into a small drilled hole, bearing in mind that the temperature indicated is that existing at the last point of junction in the couple. Other methods involve

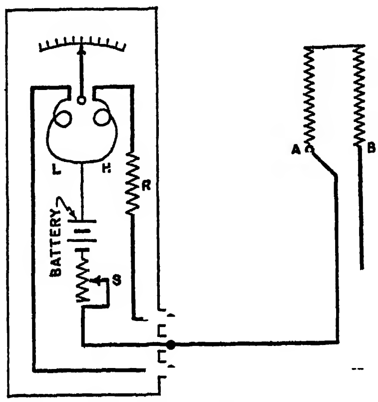

Fig. 3. Typical Resistance Thermometer Circuit and Connections

fastening the couple to the surface with adhesive cellophane, or cementing the couple with litharge in a surface scratch, and grinding it flush with the surface. Wires may also be fastened to the surface by brazing, care being taken to add as little extra metal as possible. This is a useful method for obtaining the temperature of cast-iron heating surfaces.

In any of these methods the leads should be of as fine wire as practicable, since conductance along the leads to the couple may be a source of considerable error.

Resistance thermometers depend for their operation upon the change of resistance of metal with change in temperature. Their use largely parallels that of thermocouples, although readings tend to be unstable above 950 F. Two-lead temperature elements are not recommended, since they do not permit correction for lead resistance. Three leads to each resistor are necessary to obtain consistent readings.

A typical circuit used by several manufacturers is shown in Fig. 3. In this design a differential galvanometer is used, in which coils L and H exert opposing forces on the indicating needle. Coil L is in series with the thermometer resistance AB, and coil H is in series with the constant resistance R. As the temperature falls, the resistance of AB decreases allowing more current to flow through coil L than through coil H. This

causes an increase in the force exerted by coil L, pulling the needle down to a lower reading. Likewise, as the temperature rises the resistance of AB increases, causing less current to flow through coil L than through coil H. This forces the indicating needle to a higher reading. Rheostat S must be adjusted occasionally to maintain a constant flow of current.

Instruments of this type are frequently connected through a selector switch to a number of thermal resistor elements and used to indicate temperatures in remote locations in large buildings. The direct reading feature is advantageous in this case.

Pyrometers

For measuring high temperatures, such as in furnaces, *pyrometers* are often used. Radiation pyrometers concentrate the radiant energy on a thermopile, and the reading is obtained on a galvanometer or potentiometer. Optical pyrometers require visual matching of a narrow spectral band, usually red, emitted by the object with that from a standard electric lamp.

PRESSURE MEASUREMENT

Barometer

The most accurate barometer for determining the atmospheric pressure is the mercurial type, consisting of a tube over 30 in. long closed at the top and standing in a mercury well. The barometric pressure is expressed as the height of the mercury column above the level of the mercury in the well. Such barometers are equipped with an adjustment to compensate for change in level of mercury in the well. The reading should be taken at the top of the meniscus and is obtained on a Vernier scale.

Correction for variation of the density of the mercury column and for expansion of the brass scale, which are usually calibrated for 32 F mercury and 62 F scale temperature, should be made by subtracting from the observed height in inches the value of C determined by Equation 2.

$$C = \frac{h\,(t - 28.630)}{(1.1123\,t - 10978)} \tag{2}$$

where

C = correction to be subtracted, inches of mercury.
h = observed height, inches of mercury.
t = observed temperature of the barometer, Fahrenheit degrees.

Standard atmospheric pressure at sea level is 29.921 in. Hg. Since normal atmospheric pressure decreases about 0.01 in. Hg for each 10 ft increase in elevation, it is important to make a correction if the elevation of the barometer is not that of the test apparatus. In many cases the barometric reading may be obtained from a nearby Weather Bureau Station, in which case inquiry should be made as to whether the value is as observed or corrected to sea level.

Atmospheric pressure may also be measured by an aneroid barometer which is easily portable. In this type, variations in atmospheric pressure bend the thin surface of a sealed box or tube. The aneroid type is not as accurate as the mercurial and needs frequent calibration. Most of the pressure gages used in engineering work indicate the difference between the pressure being measured and the atmospheric pressure. Such pressures are called gage pressures. Absolute pressure may be obtained by adding barometric pressure and gage pressure algebraically.

Pressure Gages

The Bourdon type gage is a widely used device for measuring pressures. The Bourdon tube is elliptical in cross-section and circular in form, and is connected by suitable linkage to a pointer which moves over a dial. An increase in pressure tends to straighten the tube and a decrease has the opposite effect. When used with high temperature steam, the tube must be protected by a water seal. When used with ammonia it must be made of steel or other material not attacked by this substance. When used for sub-atmospheric pressure, the gage is known as a vacuum gage, and is usually graduated in inches of mercury. For pressures above atmospheric, it is termed a pressure gage and is graduated in pounds per square inch. Some are made to read in both directions and are termed compound gages. Calibration is usually made by means of a dead weight tester, consisting of a platform and weights resting on a piston floating on oil. The pressure at all points in the fluid is determined from the area of the piston and the total weight resting on the oil. Adjustments

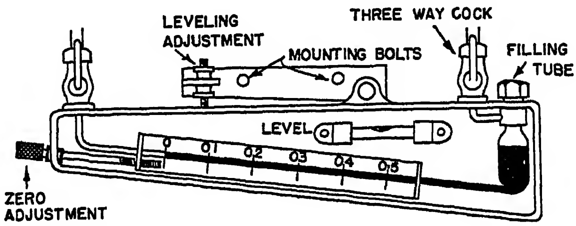

Fig. 4. Inclined Draft Gage

are provided in the gage linkage to make necessary corrections. A correction chart may be used for accurate work.

Manometers

For comparatively low gage pressures or differences in pressure between two points in a duct system, the vertical *U* tube is a simple and accurate gage and is often used for test work with various fluids such as mercury, water, kerosene, or alcohol. Readings may be in inches of any of these fluids.

For measuring pressure differences of a few inches of water, or less, *U* gages are often set at an angle for scale amplification. In commercial gages of this type, commonly termed *draft gages*, only one tube of small bore is used and the other leg is replaced by a reservoir. Although the scale is calibrated to read in inches of water, a fluid having the density and characteristics of kerosene is often used. It is necessary, in such a gage, to use a fluid having the same gravity as that for which the gage was originally calibrated, or to apply a correction if another fluid is used. Such gages may be checked one against another to detect errors in gravity of fluid. For more accurate calibration the gage may be checked against a micromanometer or a calibrating device known as a hook gage [7]. The accuracy of a draft gage is dependent on the slope of the tubes and consequently the base of the gage must be leveled carefully. It is not desirable to use a slope of less than 1 in 10.

The better grades of inclined draft gages are equipped as shown in Fig. 4. This includes a built-in level and leveling adjustment, a means of adjusting the scale to zero, and three-way vent cock connectors for checking purposes.

For measuring low pressure differences to within 0.001 in. of water very sensitive *micromanometers* are available, such as the Illinois or Wahlen, the Askania, and the Emswiler [8, 9]. Calibration of these is impossible, and readings are converted to pressure units by fundamental calculations involving the specific gravity of the fluids used and the design principles involved.

Static Holes and Tubes

In the case of low pressure air flow, the type of pressure opening used and its location are quite as important as the accuracy of the gage to

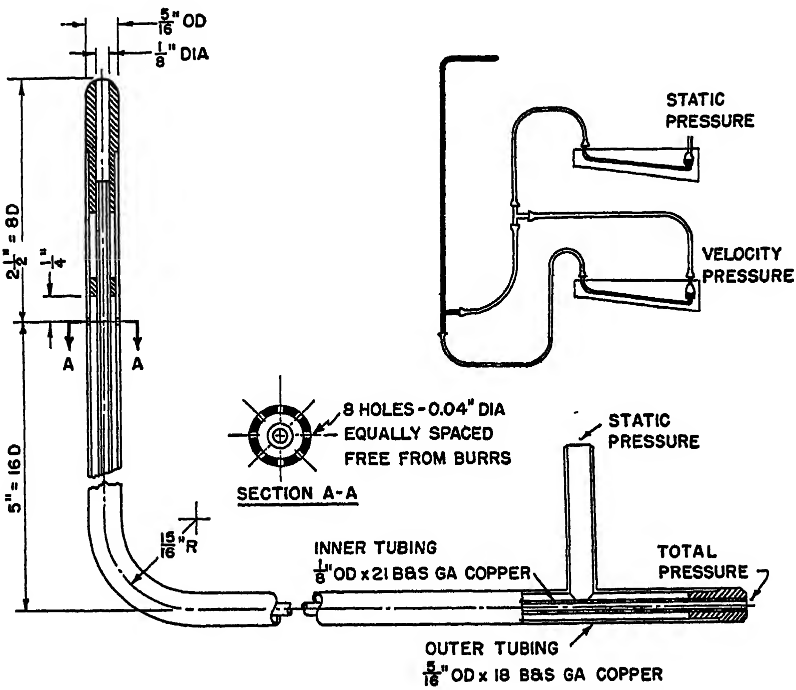

FIG. 5. STANDARD PITOT TUBE

which it is connected. Where velocities are low, as in certain plenum chambers, or where flow is free of large eddies and parallel to the walls of the conduit, a carefully drilled opening cleared of burrs and at right angles to the stream will give accurate readings [10]. The drilled depth of this hole through the wall should be at least two diameters. A second method less likely to involve error is the use of the static pressure element of a Pitot tube shown in Fig. 5. A static tube of the same general design as the Pitot may also be used, omitting the center tube. In using this instrument it should be pointed upstream, avoiding impact or eddies.

MEASUREMENT OF AIR MOVEMENT

The problem of measuring air movement may be divided into three main parts: air confined in ducts, air circulating in free spaces, and air entering or leaving such space through openings such as grilles. Other

gases may be measured by the same methods, but emphasis here is on air measurements [11].

For determining the velocity, and therefore the volume of air flowing in a duct, the Standard Pitot Tube [7] shown in Fig. 5 is probably most often used. When connected as illustrated, this instrument will give readings of both static and velocity pressure directly. From the latter the air velocity may be found from tables, or calculated from the following relation:

$$V = 1096.5 \sqrt{\frac{h_V}{d}} \tag{3}$$

where

V = velocity, feet per minute.
h_V = velocity pressure, inches of water.
d = density of air, pounds per cubic foot.

Air flow in a duct is seldom uniform. In general the velocity is lowest

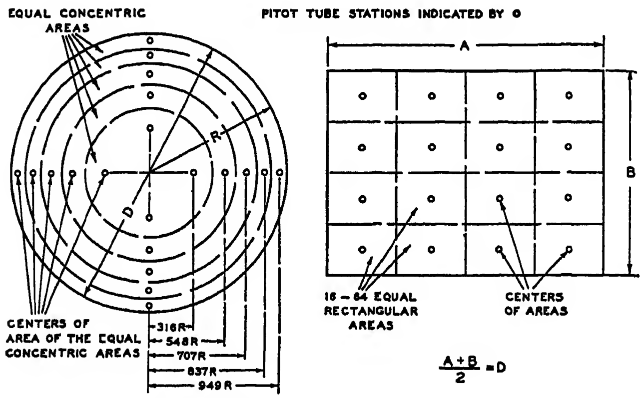

FIG. 6. PITOT TUBE TRAVERSE FOR ROUND AND RECTANGULAR DUCTS

near the edges or corners, and greatest at or near the center. For this reason a large number of readings should be taken in the manner shown in Fig. 6. In the case of round ducts not less than 20 should be taken along two diameters at centers of equal annular areas. In rectangular ducts the readings should be taken in the center of equal areas over the cross-section of the duct. The number of spaces should not be less than 16 and need not be more than 64. When less than 64 are taken the number of equal spaces should be such that the centers of the areas are not more than 6 in. apart.

In determining the average velocity in the duct from the readings given, the calculated individual velocities or the square roots of the velocity heads must be averaged. It is incorrect to use the average velocity head for this purpose.

For small pipes it is sometimes necessary to construct a Pitot tube smaller than the standard size. Such a small Pitot tube should be geometrically similar to the standard tube. Pulsating or disturbed flow will give erroneous results and every effort should be made to remove disturbances in the Pitot tube section.

The velocities used in many ducts are too low for measurement by means of pressure gages and it therefore becomes necessary to resort to other types of instruments.

Many forms of Pitot tubes other than the one described have been used and calibrated [12]. A double-ended tube [13], one end pointing down-stream, and one up-stream, is sometimes used for low velocities, but it should be carefully calibrated for accurate results. A special form of this tube design consists of two straight ⅛ in. tubes soldered together, closed at the end, and with a 0.04 in. hole in each tube opposite the line of contact. This tube is useful in exploring velocities on exhaust inlets, such as on hoods placed around grinding wheels.

The rounded approach orifice or nozzle of the general type described in the A.S.H.V.E. Unit Heater [14] and Unit Ventilator [15] Codes is an accurate air measuring device. When it is well made, the coefficient closely approaches unity. The discharge from such a nozzle is uniform [16] and provides a good location for calibration of air velocity instruments [17].

The Venturi meter is like the nozzle except for the addition of a downstream transition section that reduces the pressure drop through the measuring apparatus.

The thin-plate square-edged orifice has a coefficient of approximately 0.60. The exact value depends on the location of the connections, the pressure drop, the diameter ratio of orifice to pipe, and the sharpness of the edge [18].

Another method of air measurement uses the thermal electric principle where, by means of a measured amount of current, heat is put into the air stream. The weight of air flowing is calculated from the heat equivalent of the electrical input and the temperature rise of the air. Heat should be applied uniformly to the mass of air passing, and the small temperature rise must be determined accurately.

In certain applications the air velocity through a duct, heater coil, or heating unit may be most conveniently obtained by computation from the heat given up by the coil and the temperature rise (measured by thermocouples) of the air passing through. It is essential to have a uniform flow over the entire inlet and outlet of the heater at the plane of temperature measurement.

Air Currents in Free Spaces

One of the instruments useful in determining the velocity of air currents in free spaces is the Kata-thermometer. It is essentially an alcohol thermometer with a large bulb. The stem has two marks, one corresponding to 95 F, and the other 100 F. The instrument is heated above 100 F and then the time in seconds required for it to cool from 100 to 95 when located in the air current gives a measure of the non-directional velocity. It is important to have the Kata-thermometer dry before taking the reading. Each Kata has its own factor etched on the stem, and this factor must be used with its cooling formula or chart for obtaining the velocity. The Kata-thermometer is useful in exploring ventilated spaces to determine whether the proper air movement and distribution are being maintained. It is also used in determining the cooling power of the atmosphere, since it loses heat by radiation and convection when dry, and by radiation, convection, and evaporation when the bulb is equipped with a wetted cloth covering [19].

Another instrument for measuring low velocity air currents is the

heated thermometer anemometer [20]. This consists of a mercurial glass thermometer with a resistance winding on the bulb. Current is supplied from an external source in a measured amount. The difference between the temperature of this heated thermometer and that of an ordinary thermometer at the same location, together with the current supplied, makes it possible to calculate the non-directional velocity of the air stream.

The heated thermocouple anemometer employs a thermocouple instead of a thermometer [21].

Another instrument is the hot wire anemometer which has been made in several patterns. In general, a measured current is supplied to raise the temperature of a fine bare wire above the temperature of the surrounding air. With the use of a very fine wire, minute fluctuations in velocity may be measured, and the area exposed to radiant exchange with heated or cooled surfaces is at a minimum. This instrument is easily adapted to remote reading or recording. A group of them may be connected together to give the average velocity in a space, or the velocity at individual points within a test space, by suitable switching arrangements [22, 23].

Deflecting Vane Anemometer

The deflecting vane anemometer consists of a pivoted vane enclosed in a case, against which air exerts a pressure as it passes through the instrument from an up-stream to a down-stream opening. The movement of the vane is resisted by a hair spring and a damping magnet. The instrument gives instantaneous readings of directional velocities on an indicating scale. When used in fluctuating velocities, it is necessary to average visually the swings of the needle to obtain average velocities. This instrument is very useful for studying mixing of air in a room[24] and in locating and measuring peak velocities that may be objectionable. Various attachments are available, such as the double tube arrangement for obtaining velocities in ducts, and a device for measuring static pressures. Each instrument and the attachments for it must receive individual calibration.

Propeller or Revolving Vane Anemometer

The propeller or revolving vane anemometer consists of a light revolving wheel connected through a gear train to a set of recording dials that read the linear feet of air passing in a measured length of time. It is made in various sizes, 3 in., 4 in., and 6 in. being most common. Each instrument requires individual calibration. At low velocities the friction drag of the mechanism is considerable. In order to compensate for this, a gear train that overspeeds is commonly used. For this reason the correction is often additive at the lower range and subtractive at the upper range with the least correction in the middle range of velocities. Most of these are not sensitive enough for use below 200 fpm.

Measurement of Velocities at Inlets and Outlets of Ducts

In the field it is often desirable to make volume measurements at the face of the supply openings. It is rare to have access to the interior of duct sections where the flow is sufficiently uniform for measurement. For accuracy the instrument and its application should be checked on a similar approach and grille in the laboratory before use in the field.

Tests have shown that the propeller type anemometer can be used successfully on most of the common types of supply grilles [25, 26]. The core

area is divided into equal squares, and the anemometer is held against the face of the grille for the same length of time in each. To get the air volume in cubic feet per minute, the average corrected velocity in feet per minute thus obtained is multiplied by the average of the gross and net free area of the grille (core) in square feet.

On exhaust openings, the anemometer traverse is made as described previously. The air volume may be determined by multiplying the corrected velocity in feet per minute by the gross core area of the grille in square feet and by a coefficient for average conditions of 0.85[27].

When a propeller type anemometer is held in a stream of varying velocities, it tends to indicate higher than the true average, that is, the speed of the propeller is nearer to the top velocity in its area than it is to the minimum velocity. This is the main reason for the large difference in ratings of unit ventilators by the anemometer method and by air volume measurements in a duct approach to the inlet[28].

Any of the other anemometers described can be used within their range at the face of supply grilles when properly applied. In principle it is a case of finding the velocity at many points and using the average thus found with the correct discharge area at that cross-section. The deflecting vane anemometer equipped with a jet on the end of a rubber tube has been found especially convenient and accurate on supply grilles[29]. On modern air conditioning grilles the core area is used without a correction coefficient when the jet is held one inch away from the face of the grille. At this distance the constriction due to the thin bars has disappeared since the small air jets have reunited, and the air stream has not yet spread beyond the core dimensions. With deflecting grilles the exploring jet should be turned to the angle giving a maximum reading. With suitable traversing tips and calibration, this instrument may also be used on exhaust grilles if proper grille factors are applied[30].

While hardly a quantitative instrument, smoke is very useful in studying air streams and currents. The application of a more accurate instrument is often made more exact by a preliminary exploration with smoke. A mixture of potassium chlorate and powdered sugar in equal portions gives a very satisfactory non-irritating smoke. It is fired by a match, and since considerable heat is evolved, it should be placed in a pan away from inflammable objects.

AIR CHANGE MEASUREMENTS

Atmospheric air contains a certain amount of carbon dioxide. Its concentration is increased within enclosures by the carbon dioxide given off by occupants. The total air change through open windows, infiltration, and mechanical ventilation, may be measured by the carbon dioxide concentration but absorption of CO_2 by walls and other materials may introduce errors [31] and consequently some investigators prefer hydrogen to CO_2 for such tests. Since occupants also give off moisture, the increase in humidity may also be used as an index of ventilation within a space. Usually more direct methods of measuring air supply and air distribution are in favor.

MEASUREMENT OF RELATIVE HUMIDITY

Wet- and dry-bulb mercurial thermometers are usually used to determine relative humidity. The sling psychrometer is a common mounting of the thermometers to permit swinging. The wet-bulb wick and water

for wetting it must be clean, and the temperature of the water should preferably be slightly above the wet-bulb temperature. An air stream velocity of 900 fpm is recommended, although higher velocities will result in greater accuracy [32]. This velocity is obtained with the sling psychrometer by whirling it rapidly, followed by reading the wet-bulb thermometer quickly before its indication changes. Owing to the human element involved in this method, more accurate results are obtained with types of psychrometers in which air is drawn over the thermometers by a fan, or by a bulb operated aspirator. In ducts, the air flow itself gives the proper evaporating conditions. Several observations should be made until the minimum temperature is reached. Relative humidity may be obtained from tables or psychrometric charts [33]. Although it is common practice to use the charts which are based on a barometric pressure of 29.92 in. Hg, a correction for barometric pressure is necessary for extreme accuracy. This correction is made by multiplying the relative humidity as determined from the chart by the ratio of the observed barometric pressure and the standard barometric pressure.

For temperatures below 32 F, the water on the wick is allowed to freeze, during which time the temperature will drop below the true wet-bulb. A thin film of ice is more desirable than a thick one, and it is satisfactory to remove the wick and freeze a thin film directly on the bulb. Care must be taken to read the temperatures accurately due to the slight wet-bulb depressions. Tables for ice conditions must be used [34].

Dew-point apparatus for humidity measurements consists of a polished plated container cooled by the evaporation of a volatile liquid within. The temperature at which the first slight water vapor forms on the polished surface is the dew-point. If the temperature is below 32 F, the deposit will appear as frost. Another method of determining humidity is by chemical means in which the water vapor is removed by a drying agent and weighed on a chemical balance. A thermal conductivity method is available for temperatures above 212 F or for extremely low humidities [35].

DUST DETERMINATION

The measurement of dust is complicated by the many kinds involved. Some of the collecting methods are impingement on viscous surfaces, impingement at high velocity under water, collection on porous crucibles or filter paper through which air passes, and electric precipitation. Determination may be by direct weighing of samples or by microscopic counting. A commonly used method employs the Smith-Greenburg impinger which collects samples in water or alcohol and in which particles are counted under a microscope in various cells such as the Hatch or Dunn cells. Another method employs the Lewis [36] sampling tube with the analytical determination of the increase in weight of a porous crucible. All reports should state the method of sampling and counting. The A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work specifies the porous crucible method [37].

HEAT TRANSFER THROUGH BUILDING MATERIALS

The A.S.H.V.E. Standard Test Code for Heat Transmission Through Walls [38] describes the construction and use of the guarded hot box for determining over-all heat transmission coefficients of built-up sections including surface resistances. This apparatus consists of inner and outer insulated boxes, each having one open side which is sealed tightly against

the wall specimen to be tested. The outer box serves as a guard and a zero temperature difference is maintained between the inner box and the air space separating it from the outer box. Electric heaters are provided to maintain any desired temperature within the boxes while at the same time the outer surface of the wall specimen is subjected to a low temperature. A common method of applying this low temperature is to fit the specimen into an opening in the wall of a refrigerated room. The air temperature difference between the two sides of the specimen, and the rate of heat input to the guarded hot box are measured, from which data an over-all heat flow coefficient can be calculated.

The Nicholls heat meter is very useful for determining the heat flow through walls of buildings [39]. This apparatus consists of a guarded plate on both sides of which are mounted a series of thermocouples. The plate is calibrated so that the rate of heat flow through it can be determined by observing the temperature difference between the two surfaces. In use, the meter is clamped tightly against the surface of the wall in question, and readings of heat flow of accuracy sufficient for many test purposes may be obtained.

In June 1942, the A.S.H.V.E. adopted a standard test procedure for determining the conductivity of materials by use of the guarded hot plate [40]. This method is adapted to homogeneous materials, and conductivities obtained do not include surface coefficients. The method involved is one in which two identical samples are arranged one on each side of a heated plate having a guard and a central test section. The resulting sandwich is placed between cold plates. In operation the same temperature is maintained in the guard section and test section of the heated plate and a lower temperature is maintained in each of the cold plates from a common source of cold water. Thermal conductivity is determined by measuring the amount of heat supplied to the center or guarded portion of the hot plate, the temperatures at the hot and cold surfaces and the thickness of the specimen. The apparatus requires careful, precise construction and its operation requires care and skill. The actual design of basic types of hot plates is described in an *A.S.T.M.* publication [41].

MEASUREMENT OF HEAT EXCHANGE FOR COMFORT CONDITIONS

Several instruments have been devised to measure the effect of various factors as they relate to the comfort of the body [42]. The principal ones are the Kata-thermometer, Dufton's eupatheoscope, Vernon's globe thermometer, Winslow and Greenburg's thermo-integrator, and Yaglou's heated globe [43, 44]. These instruments were designed to obtain a quantitative measurement of the thermal exchange between the human body and its environment.

COMBUSTION ANALYSIS

The analysis of flue gases to determine completeness and efficiency of combustion is usually made chemically with an Orsat apparatus. This consists of a measuring burette, a leveling bottle, and three pipettes. Carbon dioxide is absorbed in the first pipette by potassium hydroxide, oxygen in the second by potassium pyrogallate, and carbon monoxide in the third by cuprous chloride. A known volume of gas is drawn in, and

after each of the three absorptions the reduced volume is again measured in the burette. Pressure and temperature of the gas sample are kept constant while measuring. Several passes are made through each pipette which contains tubes or glass beads to increase the wetted surface. It is essential that each reaction be completed before the next reaction is started. Since the life of the reagents is limited, it is well to keep a record of the number of samples tested. Care is needed in operation to prevent the pulling of reagents out of the pipettes into the capillary tubing and burette. Many types of recording gas analyzers are available and are usually found in the larger boiler plants.

Carbon Monoxide Measurement

A method of analyzing for low carbon monoxide concentrations completes the oxidation of the carbon monoxide in a known volume of sample, in the presence of a catalyst. The heat resulting is measured by a thermocouple calibrated in parts per 10,000 of carbon monoxide[45].

SMOKE DENSITY MEASUREMENTS

Smoke density may be judged by assigning to it the number of the Ringelmann Smoke Chart which appears to have the same color when

TABLE 1. RINGELMANN SMOKE CHART SPACINGS

Number of Card	Thickness of Lines, mm	Distance in Clear Between Lines, mm
1	1.0	9.0
2	2.3	7.7
3	3.7	6.3
4	5.5	4.5

observed at a distance of 50 ft. The charts are numbered 1 to 4 and are made of black lines cross-ruled on white as given in Table 1.

Apparatus using the photo-electric cell has been devised for recording smoke densities in large plants.

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CHAPTER 12

Physiological Principles

Chemical Vitiation of Air, Physical Impurities in Air, Thermal Interchanges Between the Body and Its Environment, High Temperature Hazards, Acclimatization, Upper Limits of Heat for Men at Work, Application of Physiologic Principles to Air Conditioning Problems, Effective Temperature Index and Comfort Zones

VENTILATION is defined in part as *the process of supplying air to, or removing air from, any space by natural or mechanical means.* The word in itself implies quantity, but air must be of the proper quality also. The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. When applied to comfort air conditioning, however, the A.S.H.V.E. Code of Minimum Requirements for Comfort Air Conditioning [1] defines it "as the process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. If an installation cannot perform all of these functions, it shall be designated by a name that describes only the function or functions performed."

CHEMICAL VITIATION OF AIR

People living indoors bring about certain physical and chemical changes in the air about them. Organic matter which is usually perceived as odors comes from the body or clothes. Moisture and heat are given off by the body. The oxygen content of the air diminishes and the carbon dioxide increases, but these changes are too slight to be significant except in air tight spaces as in submarines. There is no evidence of any toxic volatile material given off by man to the ambient air. Stale air may be offensive because of odors and may induce loss of appetite and loss of energy. Objectionable body odors have the same effects. These reasons, whether esthetic or physiological, usually make it desirable in the design of air conditioning systems to provide for the elimination or control of odors arising from occupancy, cooking, or other sources. This may be accomplished by introducing odor-free air in sufficient quantities to reduce odor concentrations by dilution to a level which is not objectionable. Odor-free air may be outdoor air or air which has been cleared of odors by sorption, washing, or other appropriate means.

In the case of vitiation by a few hazardous gases such as carbon monoxide from heating, cooking, and certain industrial processes, no satisfactory chemical treatment for the elimination of the impurity has been found. The only really satisfactory solution is elimination at the source by local exhaust ventilation; or, if this is impossible, reduction to a safe concentration by dilution. (See Chapter 10.) In the case of contamination by other matter, including volatile vapors and gases, chemical treatment for the removal or reduction of the impurities has been made available through air cleaning methods, which are discussed in Chapter 33 on Air Cleaning Devices.

When the only source of contamination is the human occupant, the minimum quantity of outdoor air needed appears to be that required to remove objectionable body odors, or tobacco smoke. The concen-

tration of body odor in a room, in turn, depends upon a number of factors, including the dietary and hygienic habits of the occupants (frequently reflecting their socio-economic status), the outdoor air supply, air space allowed per person, odor adsorbing capacity of air conditioning processes, and temperature and relative humidity. Perception of odor has been found to vary as the logarithmic function of the odor intensity, or inversely with the logarithmic function of the amount of outdoor air supplied and the air space per person.

The relation between air supply and occupancy has been reported by the Harvard School of Public Health [2] and the A.S.H.V.E. Research Laboratory [3]. The findings from the Harvard study are given in Table 1. Outdoor air requirements for removal of objectionable tobacco smoke odors are not accurately known but available information and current practice indicate the need of 15 cfm per person or more [4].

The total quantity of outside air to be circulated through an enclosure is often governed chiefly by physical considerations for controlling temperature, air distribution, and air velocity. Other factors which must be taken into consideration include the type and usage of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and the operation of the system distributing the air supply. Frequently, some of these factors, particularly the need for air movement and good distribution, may be satisfied by recirculation of inside air rather than outside air.

It will be noted that, with adequate air space, the rate of air change indicated in Table 1 is from 10 to 30 cfm per person. In rooms occupied by only a few persons such a rate of air change will be automatically attained in cold weather by normal leakage around doors and windows, and can easily be secured in warm weather by the opening of windows. With a space allotment of 400 cu ft per person, only 1½ air changes per hour are necessary to provide a ventilation rate of 10 cfm per person.

Therefore, in the ordinary dwelling with adequate cubic space allotment, no special provision for controlling chemical purity of the air is necessary (aside from removal of fumes from heating appliances). For such conditions, the control of air temperature is the major factor to be considered.

In more crowded rooms (large offices, large workrooms, auditoriums), the cubic space per person is less and it is usually impossible to admit untempered outside air without creating drafts. Here, mechanical ventilation is essential for removal of the heat and moisture produced by the occupants.

The present data regarding the effect of cubic space on ventilation requirements are not universally accepted. The Code of Minimum Requirements for Comfort Air Conditioning [1] prescribes definite minimum requirements which should be familiar to the designing engineer. It should be emphasized, however, that the code fixes *minimum*, rather than *adequate* requirements.

Notwithstanding the rapid advance made in air conditioning some persons still believe there is a stimulating quality in outdoor air (particularly country, mountain and seashore air) under ideal weather conditions, which is lacking in artificially conditioned air. It is apparent, however, that modern air conditioning insures control of the phenomena of nature for the service and comfort of man independently of weather conditions. The requirements for comfort as determined by the atmos-

TABLE 1. MINIMUM OUTDOOR AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS[5]

TYPE OF OCCUPANTS	AIR SPACE PER PERSON CU FT	OUTDOOR AIR SUPPLY CFM PER PERSON
Heating season with or without recirculation. Air not conditioned.		
Sedentary adults of average socio-economic status	100	25
Sedentary adults of average socio-economic status	200	16
Sedentary adults of average socio-economic status	300	12
Sedentary adults of average socio-economic status	500	7
Laborers	200	23
Grade school children of average class	100	29
Grade school children of average class	200	21
Grade school children of average class	300	17
Grade school children of average class	500	11
Grade school children of poor class	200	38
Grade school children of better class	200	18
Grade school children of best class	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person.		
Sedentary Adults	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person		
Sedentary Adults	200	<4

pheric environment are known and the air conditioning engineer can supply these essential requirements indoors to the same perfection as may accidentally be found at times outdoors, and keep them under control. The freedom of movement, action and thought, together with the variability of stimuli experienced by persons under ideal conditions in the country, mountains or seashore, and the psychological effect of these *wide open spaces* undoubtedly have some stimulating effect, which when compared with the monotony of confinement indoors, even in the most favorable atmospheric environment, may account for the contrast. Various experimenters have attempted to duplicate the invigorating qualities of outdoor air by the use of ozone, ionization, or ultra-violet light, but results to date have been inconclusive or negative [6].

Ozone in amounts of 0.01 to 0.05 ppm of air is allowable in comfort air conditioning. Above this limit there is a pungent, unpleasant odor and perhaps respiratory distress, depression, and stupor [7].

PHYSICAL IMPURITIES IN AIR

Dust particles of almost any type can produce irritation of the mucous membranes of the nose and throat if present in high concentrations. Certain dusts may be very harmful, but coal dust is tolerated well. The effects of various industrial dusts, pollens, etc. are discussed in Chapter 10.

A certain part of the dissemination of disease in confined spaces is caused by the emission of pathogenic organisms from infected persons. (See Chapter 13.)

While in some instances it may be possible to reduce the physical impurities of the air by dilution from a non-contaminated source, such non-contaminated sources are rarely available. Frequently the outside air contains a higher concentration of physical impurities than that within an enclosure. Therefore, it is usually desirable to reduce the concentration of physical impurities by air cleaning methods. (See Chapter 33.)

THERMAL INTERCHANGES BETWEEN THE BODY AND ITS ENVIRONMENT

Body temperature depends upon the balance between heat production and heat loss. The heat resulting from the oxidation which occurs within the body (metabolism) maintains the body temperature well above that of the surrounding air in a cool or cold environment. At the same time, heat is constantly lost from the body by radiation, convection and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss. During work the body temperature may rise. In fact, afternoon temperatures of normal persons average 1 deg above the resting value of the morning.

The fundamental thermodynamic processes concerned in heat interchanges between the body and its environment may be described by the equation:

$$M = \pm S + E \pm R \pm C \tag{1}$$

where

M = rate of metabolism, heat produced within the body
S = rate of storage, change in intrinsic body heat.
E = rate of evaporative heat loss.
R = rate of radiative heat loss or gain.
C = rate of convective heat loss or gain.

Factor M, the rate of metabolism, is always positive. The storage, S, may be either positive or negative, depending upon whether heat is being stored or depleted owing to a rise or fall in body temperature. Under ordinary circumstances (when the dew-point of the air is below the body surface temperature) the evaporation loss, E, is always positive; that is, heat from metabolism supplies this loss. R and C are positive when the surface temperature of the body is above that of the walls and air, and negative when it is below.

DuBois[8], after careful calorimeter studies on fasting, nude men, plotted the partition of body heat loss and heat production as a function of the temperature. (See Fig. 1).

Fig. 1 shows some disparity between heat production and heat loss. This disparity is S in Equation 1. In the central range of the experiments, S was quite low and no increase in heat loss by vaporization was apparent. From about 27 C to 30 C (80.6 to 86 F) a zone of easy regulation of body heat exists for nude persons which may be called a *zone of thermal neutrality* in contrast with the chilling effect noted in the cooler part of the curve, the *zone of body cooling*, and in contrast with the *zone*

of evaporative regulation at higher temperatures. In the narrow range of the neutral zone most men feel comfortable under similar conditions (nude and at rest). In this neutral zone the regulation of body temperature is accomplished by automatic variation in the blood flow to the skin, especially of the hands, feet, and head, to vary the radiation and convection losses as required to balance heat loss and heat production.

At lower temperatures throughout the *zone of body cooling* there is a progressive fall in the skin temperature, especially in feet and hands, which implies a diversion of blood from the skin into deeper organs, and a progressively greater loss of intrinsic body heat ($-S$), until the body temperature falls enough to induce shivering. The muscular activity of shivering increases heat production and so makes good the excessive heat loss, maintaining body temperature nearly normal. In cool environ-

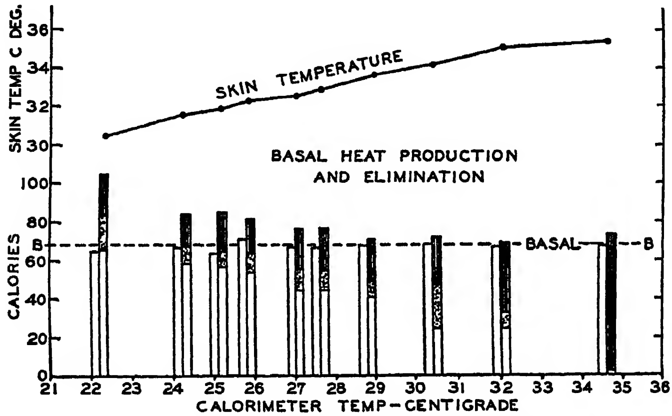

FIG. 1. HEAT LOSS FROM HUMAN BEINGS BY EVAPORATION, RADIATION, AND CONVECTION[a]

[a]Normal control, naked, in calorimeter at temperatures from 22.7 to 34 5 C First column in each experiment represents heat production as determined by indirect calorimetry, the second column, heat elimination. The portion marked with vertical lines represents vaporization, the dotted area convection, the unmarked area radiation. The skin temperature represents the average reading of 18 spots on the surface.

ments heat loss by evaporation from skin and respiratory tract remains almost constant at about 25 per cent of the heat produced in resting or slightly active subjects.

In the zone of evaporative regulation the burden of heat regulation is assumed by evaporation, radiation and convection losses already being maximal from the warm skin, flooded by maximal cutaneous blood flow. In such hot conditions the pulse rate rises. When sweat first appears it covers only part of the body, especially the head and neck, and gradually extends to drench the entire body surface [9]. All factors which affect the evaporation of water from the skin affect heat regulation in the zone of evaporative regulation. Relative humidity and air motion are most important [10]. With dry-bulb temperature above body temperature, air motion facilitates evaporative heat loss by removing hot humid air from contact with the skin and replacing it with relatively drier air.

Heat regulation in man requires an intact set of sensory nerves, a

normal sympathetic nerve supply to sweat glands and blood vessels, a great many sweat glands, and a circulatory system capable of carrying heat from muscles and viscera to the skin by circulation of the blood.

The human body possesses remarkable powers of adaptation to a narrow range of atmospheric conditions around an ideal optimum where storage is zero, and metabolism and skin and tissue temperature are at optimum values. Under these conditions, the body experiences a sensation of comfort. As skin temperature and body-tissue temperature rise or fall above or below an optimum, complex adaptive mechanisms come into play, chiefly associated with redistribution of blood supply between the skin and deeper tissues (in a cold environment) and with sweat secretion (in a hot environment). These reactions are governed by nervous or chemical stimuli from both skin and internal tissues. Nerves from the skin, for example, carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The reactions involved in cold and in hot environments are on the whole radically different in nature. The mechanisms of adjustment involved are extremely complex and, while they are receiving considerable study, a complete understanding of their operation is still lacking.

Some of the phenomena of body temperature control are shown graphically in Fig. 2. The dotted curves, from a study at the John B. Pierce Laboratory of Hygiene [11], are for subjects lightly clothed in a semi-reclining position and give the relation between the dry-bulb temperature of the environment (with about 45 per cent relative humidity) and the metabolic rate (heat production), the rate of heat dissipation by radiation and convection combined, and the latent heat loss due to evaporation from the skin and the respiratory tract. The smooth line curves from the work of the A.S.H.V.E. Research Laboratory [10] give the same relationships for healthy, male subjects (18 to 24 years of age), seated at rest and dressed in customary winter indoor clothing. The Pierce Laboratory data for the semi-reclining subjects also include the rate of heat storage (either positive or negative) due to a rise or fall in body temperature. For the normally clothed subjects, a curve gives the total heat loss (that is, the sum of the radiation, convection and evaporative losses). Here, storage is given by the difference between the metabolism and total heat loss.

The small difference between the metabolic rates for the two groups of subjects may be accounted for by the difference in activity. Heat exchange between the body and the environment by radiation and convection is greater for the lightly clothed subject, both for cool conditions where there is excessive heat loss, and for very warm conditions where there is transfer of heat from the atmosphere to the body. The two curves for evaporative loss serve to show how physiological control uses evaporation of perspiration to maintain equilibrium at high temperatures. Below about 75 F for the normally clothed subject, and below about 85 F for the lightly clothed subject, evaporation loss is minimal and constant. Above these temperatures control is obtained by the availability of perspiration for evaporation. The difference in the curves above 75 F is

probably largely determined by the difference in clothing and activity. Above temperatures from 95 to 100 F (the probable average outside surface temperature of the clothed body) the combined effect of radiation and convection causes a change from positive to negative. When the environmental temperature rises above 95 to 100 F even the greatly increased evaporative heat loss ceases to take care of the rate of heat production plus radiation and convection gains, and heat storage occurs with a consequent rise in body temperature. Above this range, even though there is inability to dissipate heat rapidly enough, metabolism actually increases. This may be accounted for by the predominance over physiological control of the purely chemical laws of increased chemical reaction with rise in temperature, and indicates the point where a

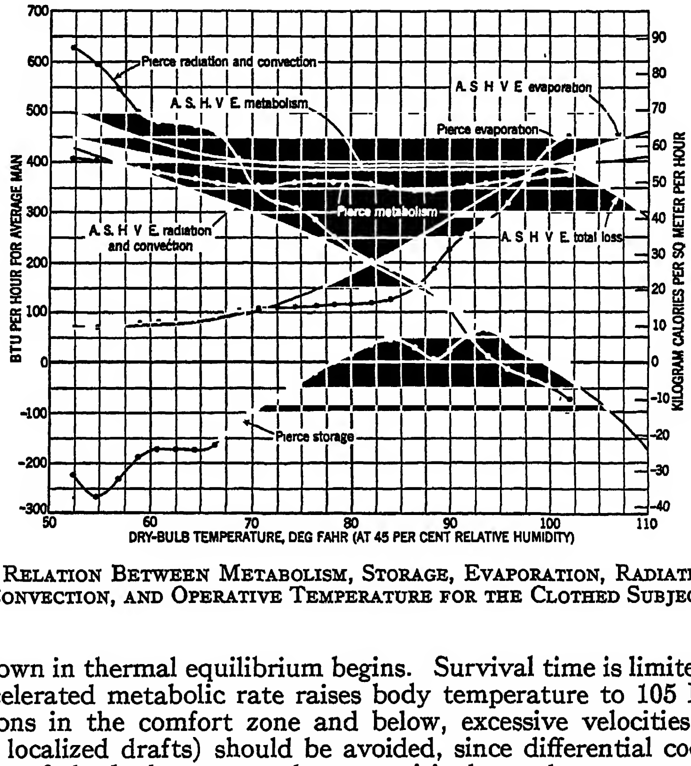

FIG. 2. RELATION BETWEEN METABOLISM, STORAGE, EVAPORATION, RADIATION PLUS CONVECTION, AND OPERATIVE TEMPERATURE FOR THE CLOTHED SUBJECT

breakdown in thermal equilibrium begins. Survival time is limited when the accelerated metabolic rate raises body temperature to 105 F. For conditions in the comfort zone and below, excessive velocities (particularly localized drafts) should be avoided, since differential cooling of one area of the body may produce surprisingly unpleasant reactions in distant areas. In one experiment [12] it was shown that the application of an ice pack to an area of 60 sq cm on the back of the neck for 15 min caused a drop of 17 deg in the skin temperature of the fingers and that this low temperature persisted for one hour after the ice pack was removed.

HIGH TEMPERATURE HAZARDS

Studies at the A.S.H.V.E. Research Laboratory [13] and elsewhere [14] during the past two decades have made available a mass of information dealing with the physiological effects of hot atmospheres on workers and means of alleviating the distress and hazards associated therewith.

Table 2 gives some of the physiological responses of men at rest and at

work to hot environments. Frequent and continued exposure of workers to hot environments results in physiological derangement affecting the leucocyte count of the blood, and other factors dealing with man's mechanism of defense against infection [15].

Wherever S (Equation 1) becomes strongly positive and body temperature rises progressively men will continue to work only for the time required for the body temperature to rise to from 101 to 103 F. When these body temperatures are exceeded men work with declining efficiency and are liable to heat exhaustion, heat cramps, or heat stroke.

Heat exhaustion is a circulatory failure in which the venous return to the heart is reduced so that fainting results [14]. Early symptoms of heat exhaustion may include fatigue, headache, dizziness when erect, loss of appetite, nausea, abdominal distress, vomiting, and shortness of breath.

TABLE 2. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp	Actual Cheek Temp (Fahr Deg)	Men at Rest			Men at Work 90,000 Ft-Lb of Work per Hour			
		Rise in Rectal Temp (Fahr Deg per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-Lb)	Rise in Body Temp (Fahr Deg per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt by Perspiration (Lb per Hr)
60	......	......	......	....	225,000	0.0	6	0.5
70	......	0 0	0	0.2	225,000	0.1	7	0.6
80	96.1	0 0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0.4	190,000	0.6	17	1.1
90	97 0	0 3	4	0.5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2 2	40	1.7	67,000	4.0[b]	103[b]	2.7[b]
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110	....	5.9[b]	137[b]	4 0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a]Data by A.S H V E. Research Laboratory

[b]Computed value from exposures lasting less than one hour.

Flushing of face and neck, pulse rate above 150, fever well above 102 F, glazed eyes, and mental disturbances as apathy, poor judgment, irritability may all precede fainting (syncope) [16]. Recovery is usually prompt when the man is removed to a cool place and kept lying down for a time, unless he has some other illness such as heart disease.

Heat cramps are painful muscle spasms in extremities, back and abdomen due, at least in part, to excessive loss of salt in sweating. Formerly common in hot industries, this manifestation of illness due to heat is now greatly reduced by use of drinking water containing 0.1 per cent salt [17], or by proper use of salt tablets. Heat cramps are readily alleviated by administration of salt solution intravenously.

Heat stroke or sun stroke is a serious effect of exposure to great heat [18]. The body temperature climbs rapidly to excessive levels often above 105 F when for unknown reasons free sweating suddenly stops. At such high temperatures coma appears and death may be imminent. Emergency measures are required to reduce the excessive body temperature by cooling quickly enough to avoid irreparable damage to the brain.

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality.

Both laboratory and field data show that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises. The incidence of minor industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature. Sickness and mortality increase progressively as the temperature rises.

ACCLIMATIZATION

When men move to deserts or to jungles some adaptation to the climate takes place. If work is gradually increased day by day and if the men can get plenty of water and salt, and get sleep each night, acclimatization will be complete in 7 to 10 days. The acclimatized man works with a lower heart rate, lower skin and rectal temperature, and more stable blood pressure than when unacclimatized [16]. The process of acclimatization requires work in the heat.

In recent tests made at the A.S.H.V.E. Research Laboratory [19], subjects were required to perform light work under very hot conditions for a 4-hr period each day. It was found that the ability of a new subject to endure these conditions showed daily improvement for a period of at least 2 weeks. However, after acclimatization was completed, a recess of several days had no effect on the endurance of the subject. Individuals differ widely in their capacity to acclimatize. Acclimatized men lose these capacities in a few weeks of temperate climate, even though they are vigorously active.

Apparently in hot dry environments acclimatization enables the body to increase the capacity to sweat and conserve salt by secreting a dilute sweat [20].

Deterioration of performance may arise during prolonged exposure to heat. Certainly when men working in hot conditions cannot get rest and sleep each day deterioration is manifest and other ill effects of heat are liable to appear.

Acclimatization to extreme conditions involves a strain upon the heat regulating system and interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature exceeding 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. Acclimatization to cold is different than to heat. Some persons regard the unnecessary endurance of cold as a virtue. They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of present knowledge of air conditioning these views are not justified. An environment averaging 64 F for the 24-hour period has been indicated as associated with minimal mortality [21].

The adaptive level changes somewhat with the season. There are also marked differences between the sexes. In the cold zone the thickness of thermal insulating tissues of women is almost double that of men, although the sensory responses to cold are similar. In the hot zone, the threshold of sweating and skin temperature levels are higher for women. The thickness and insulating value of the clothing worn are also important factors in the determination of the comfort level.

UPPER LIMITS OF HEAT FOR MEN AT WORK

In very hot conditions humidity is the limiting factor and the wet-bulb temperature assumes great importance. In 1905 Haldane recognized

TABLE 3. UPPER LIMITS OF ENVIRONMENTAL CONDITIONS FOR ACCLIMATIZED, HEALTHY, YOUNG MEN IN MILITARY SERVICE

ENVIRONMENT	REACTIONS AT THE END OF 4 HR	
	Rectal Temp F	Pulse rate
Relatively Easy	Below 101	Below 130
Difficult	101 to 102	130 to 145
Impossible	Above 102	Over 145

that 88 F wet-bulb was the limit of endurance for coal miners and later observers have concurred.

A study was made at the Armored Medical Research Laboratory [22] to determine the upper limits of environmental conditions under which a man can perform certain work. Thirteen enlisted men, who were first thoroughly acclimatized to the hot conditions, served as subjects. During each test, the subjects were required to march for 4 hr at the rate of 3 mph, carrying 20 lb packs. Tests were made under a wide range of environmental conditions, and these environments were rated in 3 zones as relatively easy, difficult, and impossible, on the basis of the physiological reactions of the subjects at the end of the 4-hr period as shown in Table 3 and Fig. 3.

Recognition of the need of air conditioning for workers in hot industries is growing rapidly and this should become an important field for the air conditioning engineer. The hot conditions may be remedied by any of the recognized comfort cooling applications. The choice of the type of system to be used in any given instance must be determined by the air conditioning engineer after a study of the conditions.

In some hot industries where few workers are engaged in spaces of large volumetric capacity the worker himself, rather than the atmosphere, can be cooled by placing him in a small cooled and ventilated booth, by

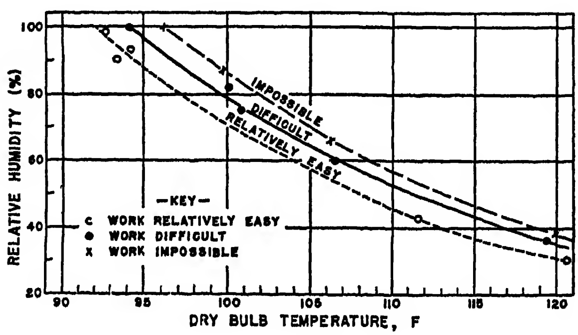

FIG. 3. THE ENDURANCE OF ENVIRONMENTAL CONDITIONS BY ACCLIMATIZED SUBJECTS WORKING AT A SPECIFIC RATE

From The Upper Limits of Environmental Heat and Humidity Tolerated by Acclimatized Men Working in Hot Environments (*Journal of Industrial Hygiene and Toxicology*, March, 1945, P. 70). Used by permission

blowing cooled air over him, or by circulating cooled air through a loose-fitting suit [23].

The A.S.H.V.E. Laboratory has studied the effects of walls of higher temperature than the air [19]. The findings are in part indicated in Fig. 4. Mean radiant temperatures were observed up to 40 deg above the dry-bulb and were found to influence physiologic processes in this region less than had been expected. For example, at 84 deg ET and 40 deg elevation in MRT 1 deg ET change is equivalent to a 4 deg rise in MRT. Similarly, at a constant ET of 90 deg and with MRT elevations of 0 and 40 deg,

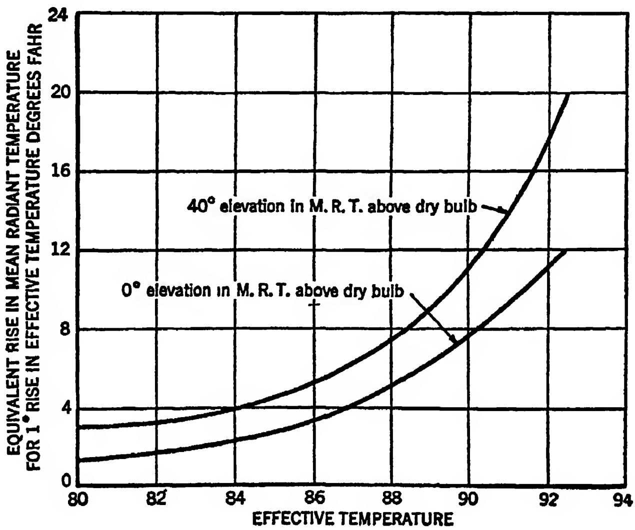

FIG. 4. EVALUATION OF EFFECT OF MRT ELEVATION IN TERMS OF EFFECTIVE TEMPERATURE

1 deg change in ET is equivalent to 7.5 deg and 11 deg rise in MRT respectively.

APPLICATION OF PHYSIOLOGIC PRINCIPLES TO AIR CONDITIONING PROBLEMS

In order to estimate cooling loads in occupied spaces it is necessary to know the metabolic rate (heat production) of man. This has been studied by investigators [24], who are agreed that it is relatively constant as a function of body surface area if determined with the subject fasting and resting quietly after a good night's sleep. The rate is high in children, and diminishes gradually with age; it increases in certain diseases and in the presence of fever. The metabolic rate is somewhat lower in women. Heat production goes up sharply with work and varies widely in different people doing the same work. The data presented in Figs. 5, 6, 7, and in Table 4 are good guides for estimation of heat production. Fig. 5 shows the total heat loss of subjects at rest and during work at different rates. Fig. 6 gives the radiation and convection losses for the same data, and Fig. 7 gives the evaporative heat loss. The data from Figs. 5, 6, and 7

permit prediction of partition of heat loss for any work rate within the usual temperature range.

EFFECTIVE TEMPERATURE INDEX AND COMFORT ZONES

The purpose of comfort air conditioning is to provide an environment in which as many persons as possible are comfortable. There is no one

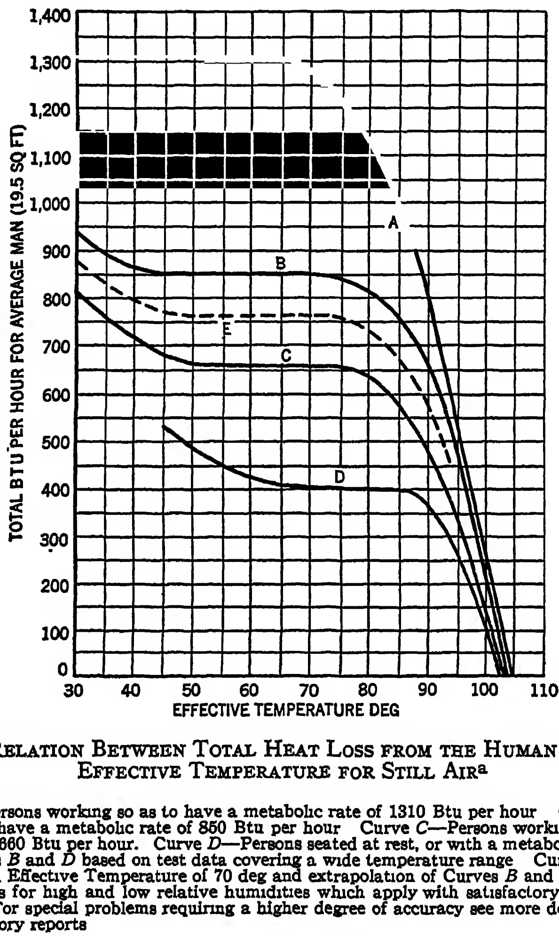

Fig. 5. Relation Between Total Heat Loss from the Human Body and Effective Temperature for Still Air[a]

[a]Curve *A*—Persons working so as to have a metabolic rate of 1310 Btu per hour Curve *B*—Persons working so as to have a metabolic rate of 850 Btu per hour Curve *C*—Persons working so as to have a metabolic rate of 660 Btu per hour. Curve *D*—Persons seated at rest, or with a metabolic rate of 400 Btu per hour Curves *B* and *D* based on test data covering a wide temperature range Curves *A* and *C* based on test data at an Effective Temperature of 70 deg and extrapolation of Curves *B* and *D* All curves are averages of values for high and low relative humidities which apply with satisfactory accuracy for most considerations. For special problems requiring a higher degree of accuracy see more detailed A.S H.V E. Research Laboratory reports

physiologic observation by which comfort can be measured. The zone of thermal neutrality differs with clothing, season, activity, and all the other factors controlling heat production (Table 5). The comfort zone is very similar to the zone of thermal neutrality [25].

Sensations of warmth or cold depend, not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer, upon air movement, and upon radiation effects. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature

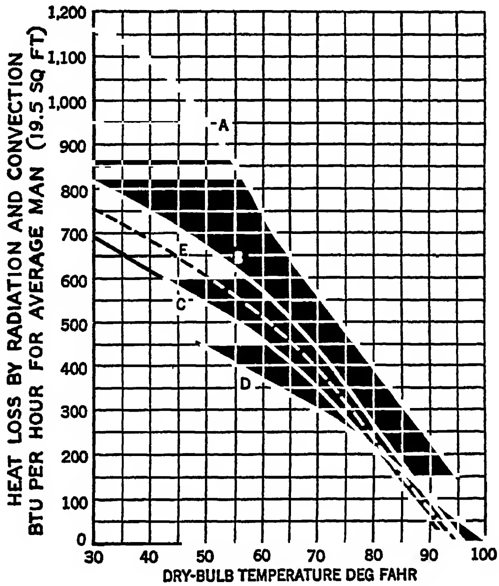

FIG. 6. RELATION BETWEEN RADIATION AND CONVECTION LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a]

[a]Loc. Cit. See footnote a, Fig. 5.

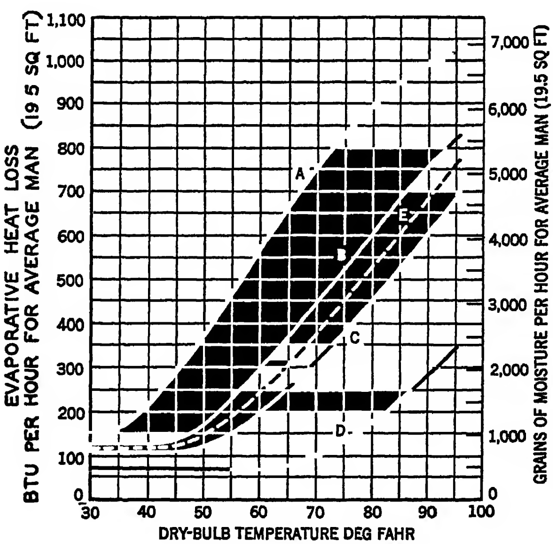

FIG. 7. EVAPORATIVE HEAT AND MOISTURE LOSS FROM THE HUMAN BODY IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Loc. Cit. See footnote a, Fig. 5.

Table 4. Metabolic Rates of Men at Different Activities[a]

Activity	Avg Total Metabolic Rate Btu per Hour
Sleeping	255
Awake, quiet	300
Seated, at rest	380
Standing, at ease	430
Walking, 2 mph	760
Walking, 4 mph	1400
Maximum Exertion	3000–4800

[a]Data were compiled from actual tests at the A S H V E. Research Laboratory and from published reports of other investigators.

with a high moisture content. Air motion makes any moderate condition feel cooler. Radiation to cold or from warm surfaces is another important factor under certain conditions.

Combinations of temperature, humidity, and air movement which induce the same feeling of warmth are called thermo-equivalent conditions. A series of studies[26] at the A.S.H.V.E. Research Laboratory established the equivalent conditions for general air conditioning work. This scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also to a considerable degree determines the physiological *effects* on the body induced by heat or cold. For this reason, it is called the *effective temperature* scale or index, and it denotes sensory heat level.

Effective temperature is an empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity, and air movement. It was determined by trained subjects who

Table 5. Comparison of Comfort Ranges With Zone of Thermal Neutrality[25]

Investigators	Effective Temperature		Operative Temp	Remarks
	Optimum Line	Range	Range	
Comfort Zone				
Houghten and Yaglou	66	63–71		Winter non-basal; at rest, normally clothed. Men and women.
Yaglou and Drinker	71	66–75		Summer non-basal; at rest and normally clothed. Men.
Yaglou	72.5	66–82		Entire year; non-basal; at rest and stripped to waist. Men.
Keeton et al	75	74–76		Entire year; basal, nude. Steady state (9 hr exposure). Men and women.
Zone of Thermal Neutrality				
DuBois and Hardy	75	73.2–76.9		Basal; nude; men.
	71.8	64.8–76.0		Basal; clothed; men.
Winslow, Herrington and Gagge			84.0–87.8	Non-basal; at rest; nude; men.
			74 –84	Non-basal; at rest; clothed; men.

compared the relative warmth of various air conditions in two adjoining conditioned rooms by passing back and forth from one room to the other.

The numerical value of the index for any given air conditions is fixed by the temperature of slowly moving (15 to 25 fpm air movement) saturated air which induces a like sensation of warmth or cold. Thus, any air condition has an effective temperature of 60 deg, when it induces a sensation of warmth like that experienced in slowly moving air at 60 deg saturated with moisture. The *effective temperature index* cannot be

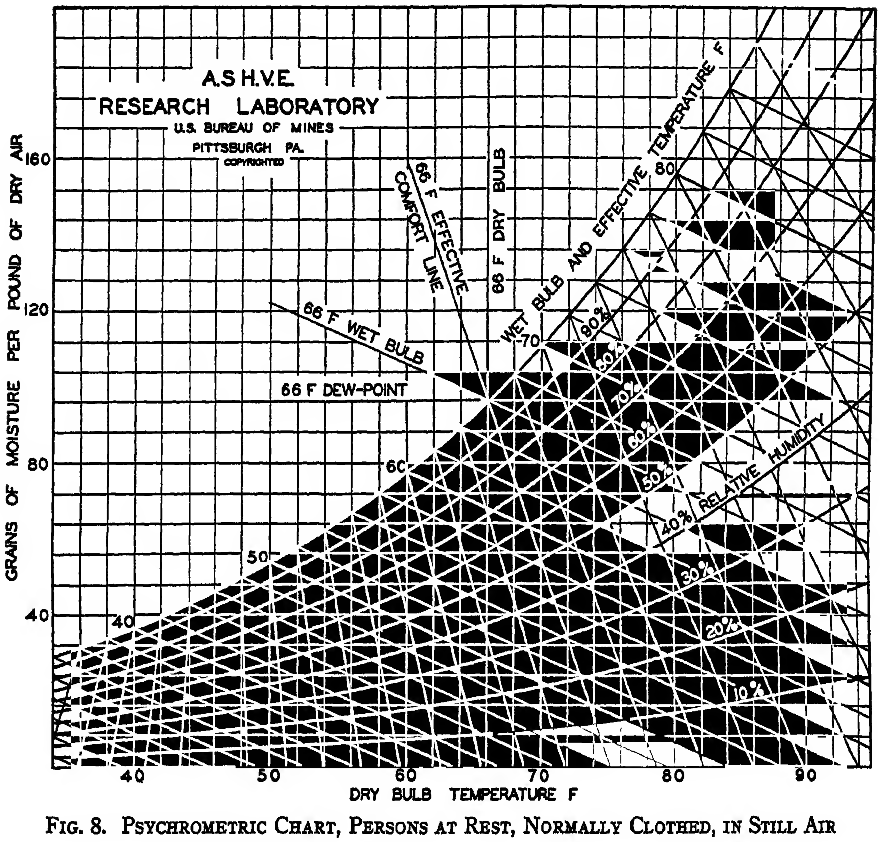

Fig. 8. Psychrometric Chart, Persons at Rest, Normally Clothed, in Still Air

measured directly but is determined from dry- and wet-bulb temperature and air motion observations by reference to an Effective Temperature Chart (see Figs. 8, 9, and 10) or tables.

Fig. 8 gives the effective temperature for any combination of dry- and wet-bulb temperatures for *still air* (15 to 25 fpm) conditions. Charts similar to Fig. 8 for air velocities of 300 and 500 fpm have been presented in some of the earlier editions of the Guide. Fig. 9 is another form of effective temperature chart embodying all three variables; dry-bulb and wet-bulb temperatures, and air velocity.

As stated previously, effective temperature is an index of the degree of warmth experienced by the body. An effective temperature line is, therefore, a line defining the various combinations of conditions which will

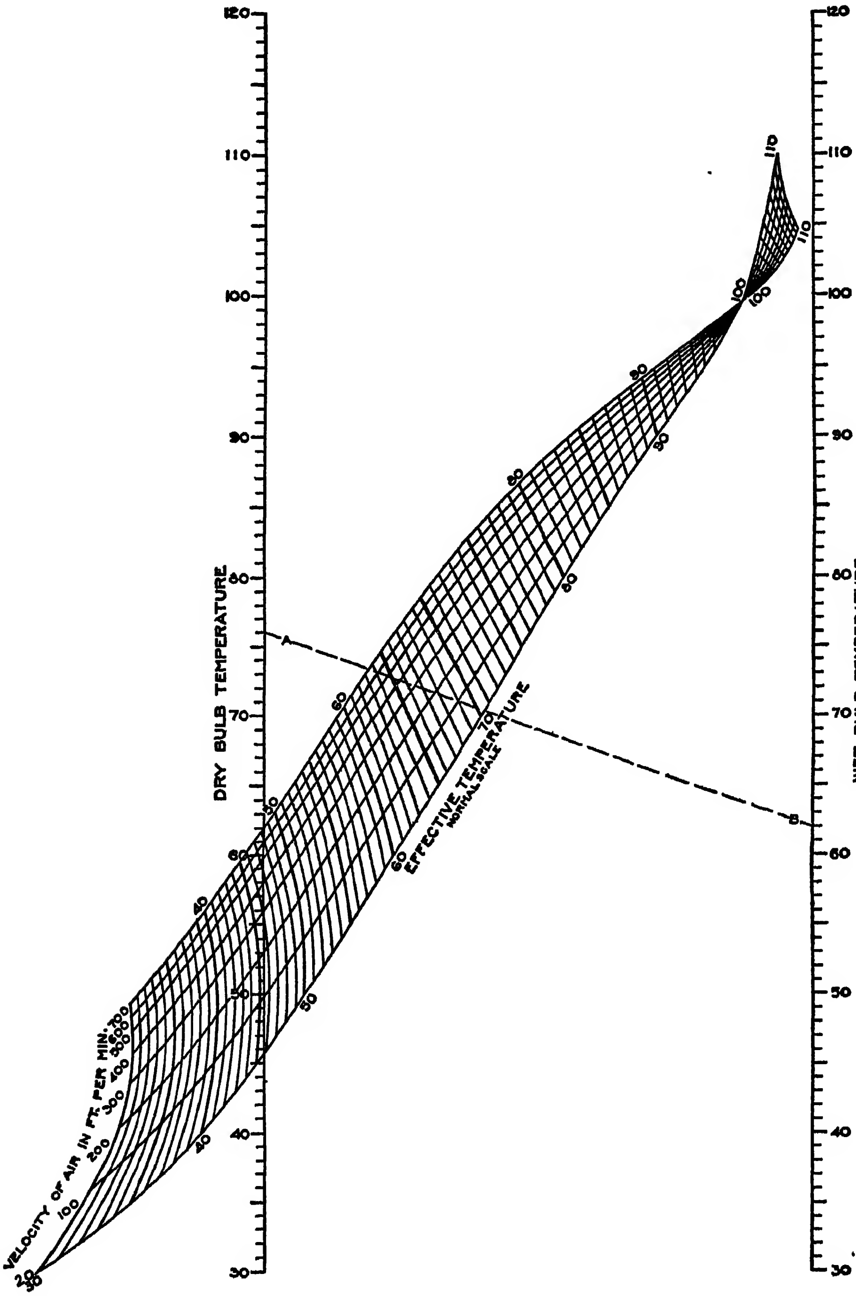

Fig. 9. Effective Temperature Chart Showing Normal Scale of Effective Temperature, Applicable to Inhabitants of the United States Under Following Conditions:

A. Clothing: Customary indoor clothing. *B. Activity:* Sedentary or light muscular work. *C. Heating Methods:* Convection type, *i e.*, warm air, direct steam or hot water radiators, plenum systems

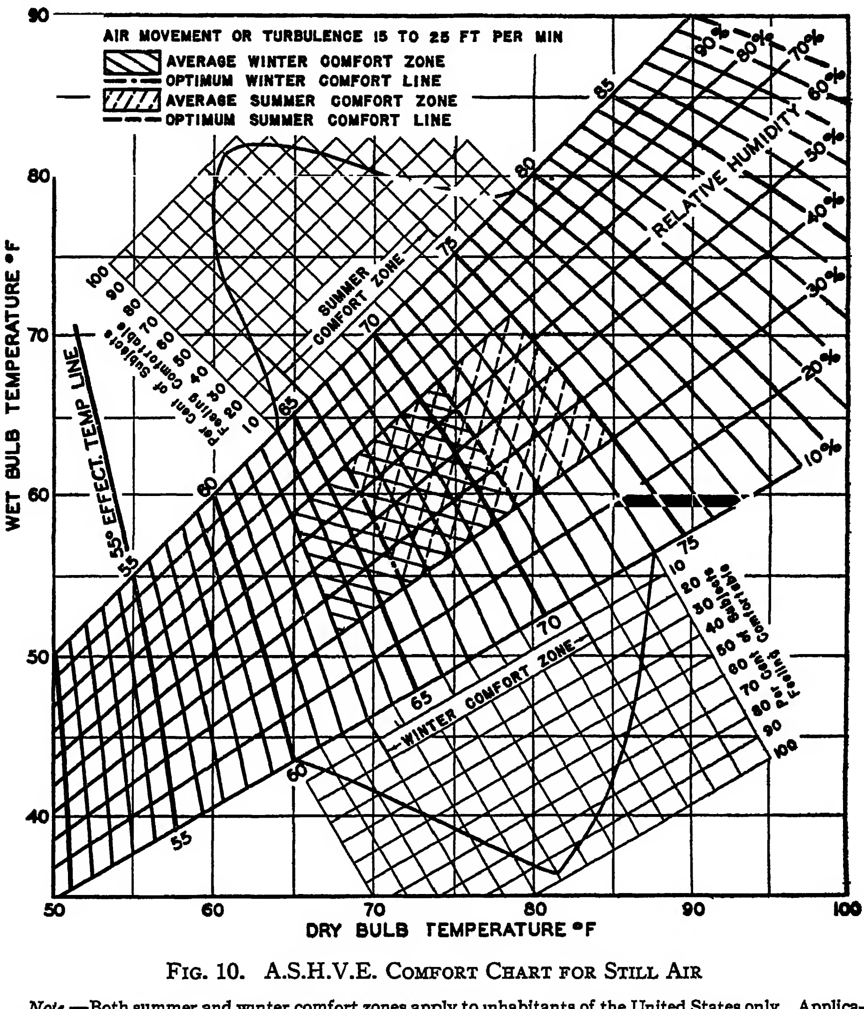

Fig. 10. A.S.H.V.E. Comfort Chart for Still Air

Note.—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type The line does not apply to rooms heated by radiant methods Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours. The optimum summer comfort line shown pertains to Pittsburgh and to other cities in the northern portion of the United States and Southern Canada, and at elevations not in excess of 1000 ft above sea level. An increase of one deg ET should be made approximately per 5 deg reduction in north latitude

induce like sensations of warmth. It does not necessarily follow that like sensations of comfort will also be experienced along the entire length of an effective temperature line. Some degree of discomfort is likely to be experienced at very high or very low relative humidities, regardless of the effective temperature. It has also been found that the optimum effective temperature varies with the season, and is lower in winter than in summer.

Fig. 10, commonly referred to as the Comfort Chart [27], is an effective temperature chart on which the summer and winter *comfort zones* have been indicated. These zones indicate the various combinations of conditions under which 50 per cent or more of the people are comfortable. Curves showing the percentage of subjects comfortable at each effective

temperature in summer and winter have been added to the chart. The summer comfort zone is indicated as extending from 66 ET to 75 ET with a maximum of 98 per cent comfortable at 71 ET. The winter comfort zone extends from 63 ET to 71 ET with a maximum of 97 per cent comfortable at 66 ET. The 71 ET and 66 ET lines are referred to, respectively, as the summer and winter comfort lines.

The comfort zones and lines as shown in Fig. 10 are based on research prior to 1932. Later studies [28] by the A.S.H.V.E. Research Laboratory indicate a desirable winter effective temperature of 67 deg, and this finding is confirmed by current practice. The shape of the curve showing the per cent of subjects comfortable in winter also justifies this conclusion, since a drop of only one degree from the 66 ET line seriously reduced the percentage of subjects comfortable.

The comfort zones in Fig. 10 are located between the 30 per cent and 70 per cent relative humidity lines. There is some evidence that the zones could be extended somewhat beyond these limitations.

It should be emphasized that a satisfactory system will not necessarily result by designing for just any combination of conditions within the boundaries of the comfort zone. As pointed out, the comfort zone covers all conditions under which 50 per cent or more of the subjects were comfortable. An air conditioning system which leaves 50 per cent of the people uncomfortable would not be acceptable. Systems should be designed to assure comfort for the maximum possible number.

The results of tests [19] made at the A S.H.V.E. Research Laboratory in very hot conditions with subjects doing light work were in very close agreement with the effective temperature chart. Other work [22] under similar environmental conditions, but with subjects walking 3 mph and carrying 20 lb packs indicated that the effective temperature lines should be more nearly horizontal. It therefore appears that the slope of the ET lines may vary depending upon the rate of work being performed.

Radiation between the occupant of an enclosure and the surfaces of the room itself and objects within the room, including windows, heating and cooling equipment, and other occupants, has an important bearing on the feeling of warmth and may alter to some measurable degree the optimum conditions for comfort previously indicated. Since the mean radiant temperature of a space is affected by cold walls and windows, as well as by the warm surfaces of heating units placed within the room or imbedded in the walls, these factors must be compensated. Likewise, in densely occupied spaces, such as classrooms, theaters and auditoriums, temperatures somewhat lower than those indicated by the comfort line may be desirable because of counter-radiation between the bodies of occupants in close proximity to each other. Such radiation will also elevate the mean radiant temperature of the room.

Many studies have been made to determine the optimum effective temperature for comfort of normal persons in both winter and summer air conditioned space, in different geographical regions and for different age groups of men and women. A group of these studies [29] was made between 1935 and 1940 by the A.S.H.V.E. Research Laboratory in Pittsburgh, and in several metropolitan districts of the United States and Canada in cooperation with the managements of offices employing large numbers of workers. Some of the results are shown in Fig. 11. Taking all of these studies together, women of all age groups studied prefer an effective temperature for comfort 1.1 deg higher than men. All men and women,

beyond the age of 40 years prefer a temperature 0.9 deg ET higher than that desired by persons below this age. The persons serving in all of these studies were representative of office workers clothed for air conditioned space in the summer season and engaged in the customary sedentary activity of office workers.

On the basis of present knowledge, for different geographical regions and age groups, the most popular temperature varies from a low of 66 deg ET for winter heating and air conditioning, to a high of 73 deg ET for summer cooling and air conditioning.

The spread for summer air conditioning for optimum confort is confined entirely to an effective temperature range of from 69 to 73 deg, and

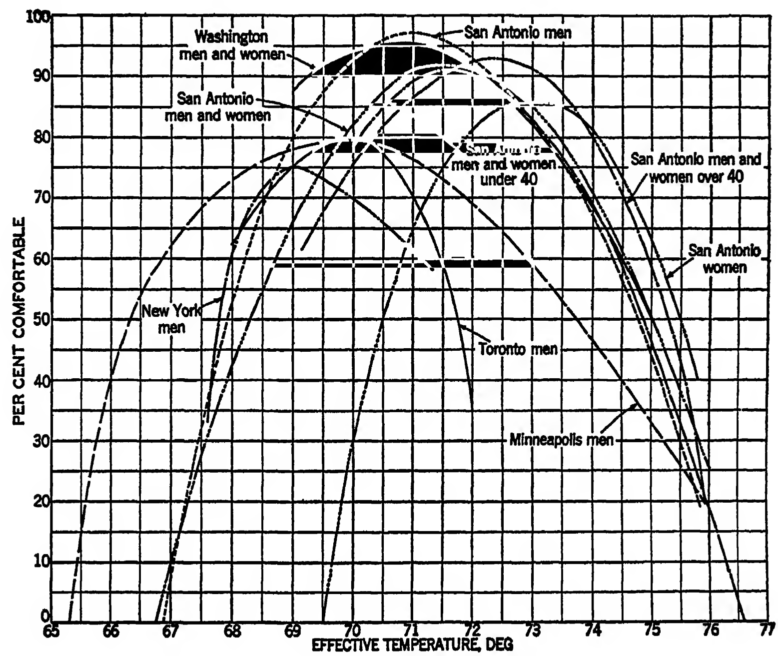

FIG. 11. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OBSERVATIONS INDICATING COMFORT

it may be presumed that for winter conditioning a like spread would exist; while for inter-seasonal conditions there will be a fluctuation between these two ranges.

Acclimatization and habits of clothing and diet account for these variations. An analyis [30] of most of the material published up to 1942, made by the A.S.H.V.E. Technical Advisory Committee on Sensations of Comfort, indicates a spread of approximately 3 deg in the optimum effective temperature for summer cooling and air conditioning due to geographical location. However, it should be recognized that variations in sensation of comfort among individuals may be greater for any given location, as shown in Fig. 11, than variations due to a difference in geographical location. The available information indicates rather clearly that changes in weather conditions over a period of a few days do not

acclimate people to different indoor conditions, but in general, people experiencing low temperatures over an extended period of time become acclimated to lower indoor temperatures, while those experiencing higher temperatures become acclimated to higher indoor temperatures.

The sudden sensation of coldness felt by persons entering a cooled and air conditioned space during the summer months, and often referred to as shock, may at times be important. It is due to the rapid evaporation of perspiration which accumulated on the skin and in the clothing during previous subjection to hot and humid outside conditions. While studies [31] have shown that for healthy individuals this *shock* is not unpleasant it is plausible, but not proven, that under some conditions it may result in unpleasant or even harmful cooling. Where a large number of occupants may enter for only a short time, 15 min or less, such occupants may be satisfied with less cooling. For long occupancy very little deviation from the optimum effective temperature is indicated.

An exit shock when leaving air conditioned space and entering a warm atmosphere is equally plausible. Experiments at the A.S.H.V.E. Research Laboratory [32] indicated no demonstrable harm to a healthy individual. Complete acclimatization occurred as soon as normal perspiration was established. Mild exercise shortened the adaptation time.

A great number of persons seem to be fairly content in summer with a higher plane of indoor temperature. Studies by the University of Illinois [33] in cooperation with the A.S.H.V.E. Committee on Research indicate that effective temperatures as high as 74.5 deg are acceptable in the living quarters of a residence, and while this condition is not representative of optimum comfort it provides sufficient relief in hot weather to be acceptable to the majority of users. It should be emphasized, however, that these are borderline cases that may be acceptable largely in the interest of economy. Comprehensive studies by the A.S.H.V.E. Research Laboratory [29] in cooperation with office staffs in widely distributed regions, including San Antonio, Minneapolis, Washington, D. C., and New York City (see Fig. 11), show conclusively that lower effective temperatures are required for optimum comfort.

The sensation of comfort, insofar as the physical environment is concerned, is not absolute but varies considerably among certain inividuals. Therefore, in applying the air conditions indicated, it should not be expected that all the occupants of a room will feel perfectly comfortable. However, when optimum comfort temperatures are applied in accordance with foregoing recommendations, the majority of the occupants should be comfortable, and it should be expected that there will be a few too warm and a few too cold. These individual differences among the minority should be counteracted by suitable clothing.

Satisfactory comfort conditions for persons at work [34] are found to vary depending upon the rate of work and the amount of clothing worn. In general, the greater the degree of activity, the lower the effective temperature necessary for optimum comfort. However, work by the A.S.H.V.E. Research Laboratory [35] indicates that under certain conditions moderate activity on the part of a person standing up and moving about may result in a slightly higher optimum effective temperature than for a person seated at rest, because of the larger body surface area exposed for heat elimination and the increase in effective air movement over his body. Where few workers occupy a large space in hot industries, work by the A.S.H.V.E. Research Laboratory [23] shows that they may be made reason-

ably comfortable by blowing relatively small volumes of slightly cooled air over them or through their clothing.

For prematurely born infants, the optimum temperature varies from 100 to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent [36]. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 to 68F with natural indoor humidities. For children (having high metabolism) at school, in winter clothes, 70 F has been considered correct, while in a gymnasium 55 F has been recommended.

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CHAPTER 13

Air Conditioning in the Prevention and Treatment of Disease

Control of Airborne Infection, Value of Air Cooling Under Tropical Conditions, Treatment of Disease, Operating Rooms, Nurseries for Premature Infants, Fever Therapy, Cold Therapy, Allergic Disorders, Oxygen Therapy, General Hospital Air Conditioning

THE late war has caused an increase of interest in the preventive aspects of air conditioning. It has re-emphasized the importance of the control of airborne infection and has demonstrated the value of air cooling under tropical conditions for the prevention of heat rash, for promoting proper rest and sleep, and in the convalescence of patients.

CONTROL OF AIRBORNE INFECTION

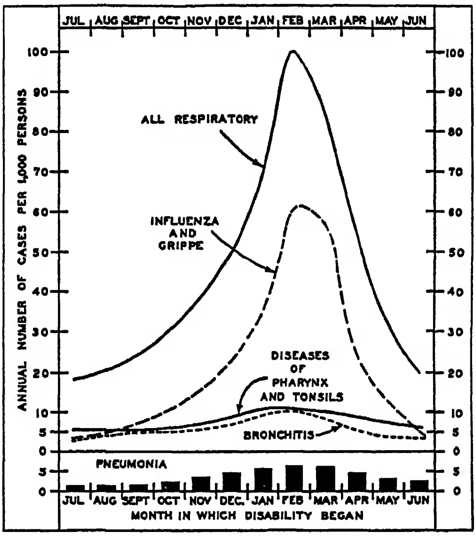
Any program of air sanitation is influenced by a number of factors[1,2,3,4]. In the winter months, the closing of doors, windows and other means of access to the outside air to conserve warmth as well as the crowding of persons indoors provides conditions conducive to a high incidence of contagion. This seasonal phenomenon, illustrated in Fig. 1 which represents a study made by the U. S. Public Health Service, will concern the ventilating engineer in so far as air quality, determined by temperature, humidity, air replenishment and type of air movement and by freedom from contamination, is a major intrinsic factor. Apart from the seasonal picture of airborne contagion are such extrinsic factors as rate of turnover of personnel and the marked susceptibility of the recruit in comparison with permanent personnel[5] as shown in Fig. 2 by studies of military personnel housed in barracks. These extraneous variables and the factor of contact infection (direct spray) tend to complicate any evaluation of the effectiveness of air sanitation for elimination of microorganisms in droplet-nuclei and droplet-dust. Thus, control measures may eliminate consistently 90 per cent of airborne organisms in laboratory tests, but cannot effect a decrease in actual incidence of infection exceeding 30 per cent. Thirty per cent may be the maximal reduction in infection possible by air treatment methods. The distinction should be clearly drawn, therefore, between the effectiveness of a procedure in laboratory tests and its effectiveness and applicability in actually reducing the incidence of airborne disease.

The following sequence of events has been postulated as occurring in a large proportion of intra-ward infections: (*a*) ejection of relatively large protected infective particles from patients, (*b*) rapid venting or settling of these particles so that those remaining airborne are in low concentration, (*c*) survival of infective particles to permit the accumulation of high concentrations on surfaces, (*d*) repeated reintroduction of infective particles into the air under the stimulus of ward activities or by air currents of the order of 50 fpm over the floor, and (*e*) extension of infective areas by air turbulence throughout the ward or hospital. The most important link in this probable infection chain has been demonstrated to be the reintroduction of particles into the air[6].

Intensive studies on air disinfection have indicated two distinct control measures, (*a*) suppression of dust and lint, and (*b*) disinfection of droplet-nuclei.

Well controlled, large scale tests of the various methods of air sterilization conducted in barracks [7, 8] have confirmed the importance of dust control in minimizing the spread of airborne disease, a consideration which has guided the practices of ventilating engineers for a number of years. The importance of the dust factor has been emphasized by many engineers and has been convincingly demonstrated by subsequent bacteriologic studies aboard ships.

Treatment of floors and bedclothes with oil emulsions has proved effective in reducing bacterial dispersion by as much as 90 per cent in



Occurrence of diseases causing disability for 8 consecutive days or longer in a group of 100,000 wage earners (10 per cent women) in different industries.

[a]Graph obtained from Dean K. Brundage, U. S. Public Health Service.

FIG. 1. STUDY OF AVERAGE MONTHLY FREQUENCY (1921-1926 INCLUSIVE) OF SPECIFIED RESPIRATORY DISEASES[a]

Army barracks and station hospitals [8]. The incidence of acute respiratory infections was from 10 to 30 per cent lower in barracks with oiled floors and bedclothes than it was in control barracks which received no special treatment.

No simple method for disinfecting droplet-nuclei has yet been devised. Under favorable laboratory conditions, propylene glycol in concentrations of 0.07 to 0.14 milligrams per liter, and triethylene glycol in a concentration of 0.0045 milligrams per liter were highly germicidal for most airborne bacteria in clean air when the relative humidity was between 40 and 60 per cent [9, 10, 11]. Under practical conditions, however, particularly in the presence of dust in the air, glycol effectiveness is much reduced. The use of other chemical aerosols that have been tried is limited by their toxicity, odor, or destructiveness to fabrics and metals.

Ultraviolet radiation of floors and upper air has been studied exten-

sively at the Naval Training Center, Sampson, N. Y. In barracks housing naval recruits, hospital admissions for respiratory infections (mostly *catarrhal fever*) were 25 per cent lower in a group of men exposed to ultraviolet radiation—(2537 Angstrom Units, 1 to 7 ergs per (cm^2) (sec) at bed level)—than they were in adjacent control barracks without ultraviolet radiation[12]. A combination of ultraviolet radiation and dust control measures is believed to be more effective than when either one of the two is used alone, but the proof for this has yet to come.

VALUE OF AIR COOLING UNDER TROPICAL CONDITIONS

The commissioning of a class of naval hospital ships with all wards, laboratories and living spaces air cooled is a notable achievement to provide better treatment of patients, especially those suffering from extensive burns, by control of environmental factors. Although statistics

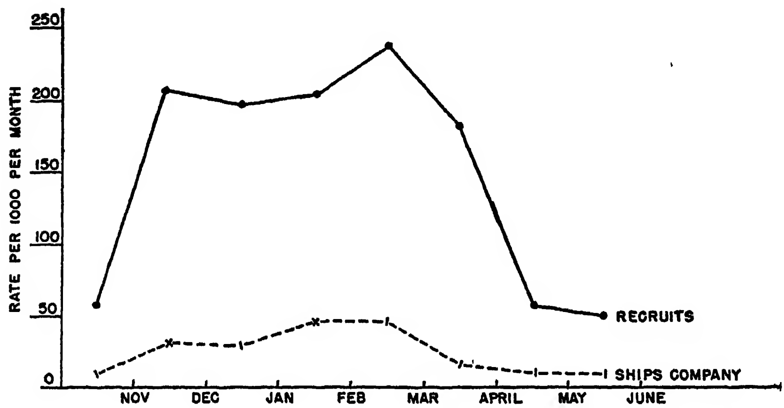


Fig. 2. Monthly Incidence of Acute Respiratory Illness Among Naval Recruits and Ship's Company (Permanent Personnel) [5]

are not at hand to indicate the deaths or retarded recoveries of patients due to lack of air cooling in ships operating in tropical waters, it is generally agreed among competent observers that high temperature and humidity are major factors in prolonging disability and increasing mortality of the sick and injured. Physiologic data obtained on healthy men, moreover, show the large loss of body fluids and the stress on the cardiovascular system in terms of increased pulse rate when these men are continuously subjected to high temperatures. Even at rest about 50 cc of fluid per hour are lost as sweat [13] through intact skin. In burn patients the difficulty, encountered in temperate climates, of maintaining fluid and electrolyte balance is tremendously augmented by the additional evaporative fluid loss in hot environments.

Frequently from 50 to 75 per cent of personnel aboard naval vessels operating in tropical waters are afflicted with heat rash to a degree that interferes with rest and sleep. In carefully controlled experiments [13] it was possible to produce a fulminating type of rash in all men living continuously at an effective temperature of 85 (90 F dry-bulb and 83 F wet-bulb). In the control group, 12 out of 24 hr were spent in a relatively cool atmosphere of 75 ET (80 F dry-bulb, and 70 F wet-bulb). These

men either remained free from heat rash or occasionally developed a mild form. Thus, intermittent cooling to a degree which prevented sweating in men at rest eliminated a serious handicap to good performance of duty.

In both laboratory tests and aboard hospital ships a relatively cool living environment of 76 to 78 ET provided an atmosphere conducive to rest and sleep without sweating. Berthing spaces tended to have extremely low odor levels. Motivation, initiative and alertness, in contrast to the usual irritability and lack of incentive incident to residence in tropical climate, were maintained [14, 15, 16].

Little has been done, however, to obtain practical methods for application of air conditioning under heavy heat loads and on the enormous scale that would be needed to modify life in the tropics. It is not improbable that cooled houses in a tropical climate, if used consistently for one generation, might modify the whole character of a population [17]. The obvious advantages of part time cooling on personnel to promote rest and sleep in tropical areas would provide a prophylactic measure of great potential importance.

TREATMENT OF DISEASE

In the past few years considerable progress has been made in using air conditioning as an adjunct in the treatment of various diseases. Among the important applications are those in operating rooms, nurseries for premature infants, maternity and delivery rooms, children's wards, clinics for arthritic patients, heat therapy, cold therapy, oxygen therapy, X-ray rooms, the control of allergic disorders, and for the physiological effects in industry.

OPERATING ROOMS

The widest application of air conditioning in hospitals is in operating rooms. Complete air conditioning of operating wards is important because winter humidification helps reduce the danger of anesthetic gases; summer cooling with some dehumidification is needed to eliminate excessive fatigue and to protect the patient and operating personnel; and finally, filtering aids the removal of allergens from the operating room air.

Reducing Explosion Hazard

Explosion hazards in operating rooms began with the introduction of modern anesthetic gases and apparatus. Ether administered by the old drop method gives rise to an explosive mixture, but in practice this method is still regarded as comparatively safe. When ether is mixed with pure oxygen, or nitrous oxide in certain concentrations, the explosion hazard may be as great as with ethylene-oxygen, or cyclopropane-oxygen mixtures [18].

Of the anesthetic gases nitrous oxide alone does not explode but supports combustion. Ether, vinyl ether, ethylene, and cyclopropane are as potentially dangerous as gasoline or illuminating gas in the home [19]. Chloroform does not explode violently in contact with flame but decomposes to liberate phosgene. All of the anesthetic gases and vapors except ethylene are heavier than air. Although the incidence of injury or death from explosion is negligible compared with other hazards in the operating room, the dramatic features surrounding an explosion justify continued investigation to eliminate the hazard.

During the course of ethylene anesthesia, the mixture, usually 80

per cent ethylene and 20 per cent oxygen, is so rich that the danger of explosion is slight in the immediate vicinity of the face mask, but leakage of ethylene into the air may accumulate to any lower concentration, and thus introduce a serious hazard. The most dangerous period is at the end of the operation when the patient's lungs and the anesthesia apparatus are customarily *washed out* with oxygen with or without the addition of carbon dioxide. Even when this procedure is omitted, it is difficult in practice to avoid dilution of the anesthetic gas with air during the normal course of breathing following the administration. In either case the mixture would pass through the explosion range and extraordinary precaution is necessary for the safety of the patient and operating personnel.

In a study [20] of 230 anesthetic explosions and fires, 70 per cent of the explosions and 60 per cent of the deaths were caused by igniting agents other then static sparks. In 1941 the *National Fire Protection Association* [21] made certain recommendations for safe practice based on available information. Some of these recommendations are:

Windows should be kept closed so that the air conditioning system can prevent pooling of explosive anesthetic gases. Twelve air changes per hour and a humidity of 55 per cent are advised. If a higher humidity were compatible with the well being of the patient and personnel, it should be maintained. All electrical installations should comply with the standards set by the *National Electrical Code* for use in explosive situations. Cautery equipment should not be used in hazardous locations. To prevent static sparks, all bodies in an operating room should be conductive or coupled. It is essential that adequate grounding be provided for the floor and every object in the operating room. Conductive rubber should be used on shoes, leg tips, operating table coverings and all rubber parts of the anesthesia equipment. All furniture in contact with the floor should be metal. In the absence of complete grounding facilities, the simple method of intercoupling patient, operating table, anesthetist and gas machine at ground potential may be used.

Experience has shown that neither high humidity nor intercoupling devices have eliminated the danger from static electric discharge. The removal of gas concentrations from the operating table area by means of specially devised exhaust ventilation should be thoroughly tested. Portable duct systems as installed aboard ship should be acceptable. Serious explosions can occur in a closed system but proper precautions will reduce this hazard to a minimum.

It should be realized that when a room and the occupants have been completely grounded there is always the possibility that the patient or the operator might receive a dangerous shock if a short circuit developed in any of the electrical equipment.

A comprehensive study of the explosion problem and of the general causes and prevention of operating room hazards by the *University of Pittsburgh*, the A.S.H.V.E. Research Laboratory, and the *U. S. Bureau of Mines* has led to a fruitful attempt to eliminate the explosive range of cyclopropane, one of the best but most difficult gases to handle. The use of helium as a diluent in the total gaseous mixture controls the oxygen concentration by displacement and, because of its flame quenching properties, it is the ideal gas for this purpose. In addition, a gaseous mixture containing helium is more difficult to ignite by electric discharges and this quality also increases the safety factor of anesthetic administration.

Operating Room Conditions

Little is known about optimum air conditions for maintaining normal body temperatures during anesthesia and the immediate post-operative period. An anesthetized patient displays dilation of blood vessels in the skin resulting in profuse sweating and (it has been believed) inability to regulate body temperature. From this it was concluded that all anesthetized patients suffered considerable heat loss, although there may be little more than 0.8 F variation in the rectal temperature during the course of the operation[22]. The severe physiological effects, such as excessive sweating and rapid pulse, of high operating room temperatures on attendants and patients during the hot months signify the need for proper cooling. A comparison of surgeons' statements who operate in both air conditioned and non-air conditioned rooms strongly indicates that the recuperative power of the patient is greater when operated upon in air conditioned rooms[22].

Although the comfortable air conditions for the operators are not identical with those for the patient, it is usually not difficult to compromise within a range of 55 to 60 per cent relative humidity and 72 to 80 F temperature. The work just cited reported that 68 to 70 F effective temperature not only furnished comfort for the operating room workers, but apparently prevented exhaustion of the patient as evidenced by rapid convalescence in the recovery ward. Additional heat may be furnished to the patient locally or by suitable covering according to body temperature in individual cases.

In the control of airborne infection in the operating room the prevention of dispersal of infectious materials into the air, control of dust and proper ventilation supersede attempts to remove or kill pathogenic organisms.

In an investigation recently conducted at the University of Pittsburgh, in a cooperative research program with the Society, comparative studies were made on bacterial content of conditioned and non-conditioned operating rooms. From these studies[23] it was concluded that the bacterial content of conditioned operating rooms was considerably less than that of non-conditioned rooms.

Bacterial counts aboard an air conditioned submarine were found to be exceptionally low and not cumulative with time although all of the air was recirculated for more than 12 hours[24] without replenishment. The removal of bacteria by the process of air cooling and condensation of moisture out of air merits further study[25].

The degree of air contamination can be reduced by proper ventilation if velocity of air over the floor does not exceed 50 fpm. Research is in progress on the use of filtered air flowing through a system of mechanical cleaners which protect the patient against infection from attendants and from bacteria-containing air in the corridor or ward[26].

Operations may be postponed on allergic patients during asthmatic manifestations through fear of complications. The removal of air-borne allergens, therefore, is in some cases an important function of the air conditioning system in preparing patients for operation.

Central system air conditioning plants and unit air conditioners prove satisfactory in operating rooms when producing between 8 and 12 air changes per hour of filtered and properly conditioned air without recirculation during the course of anesthesia. A separate exhaust fan system is usually necessary to confine and remove the gases and odors. Double windows are desirable and often necessary to prevent condensation and

frosting on the glass in cold weather and to minimize drafts. The air flow of 8 to 12 air changes in operating rooms should: (1) reduce the concentration of the anesthetic to well below the pharmacologic threshold in the vicinity of the operating personnel, (2) remove the great amounts of heat and sometimes moisture, from sterilizing equipment if inside the operating room, from the powerful surgical lights, from solar heat, and from the bodies of the operatives, and (3) provide extra capacity for quickly preparing the room for emergency operations. Much can be gained by thermal insulation of sterilizing equipment and by thorough exhaust ventilation of sterilizing rooms adjoining the operating rooms.

Too great a difference in temperature between the operating room and the final hospital destination of the patient, including corridors and elevators, is conducive to infections of the upper part of the respiratory tract and post-operative pneumonia. A suggested remedy is a recovery ward in which conditions closely approximate those of the operating room and in which the patients remain from one to four days. Satisfactory conditions in the recovery ward not only hasten convalescence, but dispel the fear frequently found in patients who must undergo operations during the hot seasons [27].

Experience has shown that a few hours after the operation the temperature of the post-operative room can be decreased a few degrees below that of an overheated operating room to stimulate recovery.

NURSERIES FOR PREMATURE INFANTS

One of the most important requirements in the care of premature infants is the stabilization of body temperature. This is necessary because their heat regulating systems are not fully developed; the metabolism is low and the infants generally exhibit marked inability to maintain normal body temperatures. The resistance to infection is low and mortality rate high.

Air Conditioning Requirements

The optimum air conditions for growth and development of premature infants were determined by extensive research [28] at the Children's Hospital, Boston, Mass., using four valid criteria, namely, stability of body temperature, gain in weight, incidence of digestive syndromes, and mortality. Individual temperature requirements varied widely (from 72 to 100 F) according to the constitutional state of the infants and body weights. The optimum relative humidity was about 65 per cent, and the air movement less than 20 fpm.

A single nursery conditioned to 77 F and 65 per cent relative humidity was found to fulfill satisfactorily the requirements of the majority of premature infants. Additional heat for weak (or debilitated) infants may be furnished in the cribs or by means of electric incubators placed inside the conditioned nursery, and the temperature adjusted according to individual requirements. In this way multiplicity of chambers and of air conditioning apparatus is obviated; the infants in the heated beds derive the benefit of breathing cool humid air, and the nurses and doctors need not expose themselves to extreme conditions.

Importance of Humidity: Although external heat is an important factor in the maintenance of normal body temperature, humidity appears to be of equal or greater importance. When the premature nurseries at the Children's Hospital were kept at relative humidity between 25 and 50

per cent for two weeks or longer, the body temperature became unstable, gain in weight diminished, the incidence of gastro-intestinal disturbances increased, and the mortality rose. On the other hand, continuous exposure to air conditions with 55 to 65 per cent relative humidity gave satisfactory results over a period of years. The initial physiologic loss of body weight (loss occurring within first four days of life) was found to vary inversely with the humidity. In the old nurseries with natural humidity it averaged 12.4 per cent of the birth weight; in the conditioned nurseries it was 8.9 per cent with 25 to 49 per cent relative humidity, and 6.0 per cent with 50 to 75 per cent relative humidity. The number of days required to regain the birth weight was correspondingly maximum in the old nursery and minimum in the conditioned nurseries under high humidity.

Maximum gains in body weight occurred in the conditioned nurseries under high humidity (55 to 65 per cent) in infants weighing less than 5 lb. The gains were less under low humidity (25 to 50 per cent) in the same nurseries, and in the old nurseries prior to the installation of air conditioning apparatus.

The incidence and severity of digestive syndromes, with diarrhea, persistent vomiting, diminishing gain or loss of body weight, and other symptoms, were generally from two to three times as high under low as under high humidity.

Summarizing, the best chances for life in premature infants are created by maintaining a relative humidity of 65 per cent in the nursery and by providing a uniform environmental temperature just sufficiently high to keep the body temperature within normal limits. Medical and nursing care are, of course, factors of equal and sometimes of greater importance.

Air Conditioning Equipment

Many of the installations now in use are of the central system type providing for filtration, for humidification and heating in cold weather, and for cooling and dehumidification in hot weather. A ventilation rate, between 8 and 12 air changes, is desirable to remove odors and maintain uniformity of temperatures in extremes of weather. Recirculation should not be used in these wards owing to odors and the possibility of infection. There should be a frequent change in spray water.

Control of Airborne Infection

The protection of the premature and older infant against infection is of the utmost importance. It was found in one installation equipped with air conditioning, germicidal lights and mechanical barriers that air conditioning alone did not prevent the spread of respiratory cross-infections. Bactericidal ultraviolet light barriers and air conditioning or mechanical barriers and air conditioning were efficient[29].

FEVER THERAPY

Artificial production of fever in man is an imitation of nature's way of overcoming invading pathogenic organisms. The action may be direct and specific by destruction of the invading organism within the safe limit of human temperatures, or indirect in the case of heat resistant organisms, by general mobilization of the defensive mechanisms of the body, which retard or neutralize the activity of pathogenic bacteria and

their toxins.

Although the action may be direct and specific by destruction of the invading organisms within the safe human limits, fever therapy exerts much of its benefit through the improvement of the mechanism of bodily defense. A serious challenge to the theory on which fever therapy is based comes from the demonstration that high fever causes a reduction in the concentration of circulating antibodies in experimental animals. Clinically, it has been shown that there is no change in the per cent of phagocytes which engulf bacteria in patients during fever therapy, although the action of the complement fixing antibodies may be temporarily diminished.

Patients for fever therapy should be carefully selected. The most serious complications which may arise are heat stroke, heat exhaustion and circulatory collapse. The chief minor complications are spasm, heat cramps, fever blisters and mild dehydration.

The limits of induced systemic fever are usually between 104 and 107 F (rectal), and the duration from 3 to 8 hours at a time. The total period of fever treatment varies with the type of the organism involved from a few hours to 50 or more.

The diseases which respond favorably to artificial fever therapy are gonorrhea and its complications (which include arthritis, pelvic infections in women, and involvement of the eye), syphilis, chorea, infectious arthritis (non-gonorrheal), encephalitis, and some forms of asthma. There are other conditions which show promise under this treatment; but the most striking results are seen in gonorrhea and syphilis, since the causative organisms can be destroyed at temperatures compatible with human life.

Equipment for Production of Fever

Artificial fever can be induced by injections of various crystalloid or colloid substances, bacterial products of typhoid and malarial organisms, or by physical methods using hot baths, radiant heat cabinets, hot humidified air cabinets, or by short wave diathermy in combination with a cabinet.

The relative advantages of various methods have been evaluated clinically [30, 31]. Among the devices for the production of fever by physical means, the one most widely used is the hot humid air or *air conditioned* cabinet. This apparatus was developed at the Kettering Institute for Medical Research at Miami Valley Hospital in Dayton, Ohio.

In the earlier studies of the Society [32], temperatures were elevated more easily using saturated atmospheres. A fever therapy apparatus [33] using these same principles has proved efficient as a means of inducing and maintaining fever in a body with small likelihood of burns because of the comparatively low dry-bulb temperatures.

When heat is necessary in treating legs or arms, such media as short or long wave diathermy, infrared, water baths, etc. have been used extensively. A recent development, a saturated atmosphere heating unit, similar to one previously described has proven satisfactory, because heat may be administered over longer periods which render deep heating possible without fear of burns or shocks. Local heating has been somewhat satisfactory in relieving the painful symptoms of peripheral vascular disease.

Some investigators employ short wave diathermy within the cabinet during the induction phase. When the desired body temperature has

been reached by electrical induction, the atmosphere of the enclosure is kept at saturation to prevent heat loss, thus maintaining the patient's temperature at the desired point. The two underlying principles in the production of fever by the hot, humid air cabinet are: (1) the transfer of heat by conduction from the circulating hot air to the body and (2) prevention of heat loss. The latter is more important. In an atmosphere of high humidity, the heat loss by evaporation is markedly decreased.

The chief physiologic effects exhibited during fever therapy are: (*a*) the cardio-vascular system is stimulated to increase the amount of blood to various parts of the body, (*b*) a mobilization of blood elements by the blood forming system, (*c*) general metabolism is increased, (*d*) only slight changes in blood chemistry except that the chlorides may be depleted if dehydration occurs.

The therapeutic value of fever therapy for certain diseases has been definitely proved. The combination of chemo-therapy and fever has proved to be more effective than when either is employed separately. This is especially true in regard to gonorrhea and early syphilis.

COLD THERAPY

In contrast to fever therapy the use of cold as a means of anesthesia and treatment is of established importance. Refrigeration anesthesia is being more widely used and the advantages of the method claimed by Allen have been completely confirmed [34].

It has been demonstrated that the cooling of limbs and other parts with ice-water or ice, cracked or pulverized, down to near the freezing point (5 C or 40 F) is harmless. Freezing must be avoided. There is a temporary retardation or suspension of life, with resumption of cellular activity as the temperature returns to normal. A human limb can remain bloodless and anesthetic below a tourniquet for at least 8 hr and perhaps up to 48 hr without injury, while the rest of the body remains warm. Where amputation is indicated it can thus be done without pain, loss of blood or strength and also without shock. There is no apparent interference with the subsequent healing of the stump. Refrigeration anesthesia is of importance not only in amputation but also in the control of hemorrhage, pain, infection and shock during the transportation of patients with traumatized limbs.

Apart from the advantages of cold in amputations, cryotherapy has been beneficial in the treatment of burns and arterial obstruction, and essential in the treatment of frostbite and immersion foot. It has been used successfully for dental anesthesia. The principle of hibernation in which the body as a whole is cooled for as much as five days in air temperatures between 50 to 60 F and applied to such conditions as morphine addiction, leukemia, and schizophrenia continues to be experimental. Prolonged cold therapy which keeps the body temperature below 95 F depresses the vital processes and is fraught with danger.

The methods used for refrigeration depending upon available facilities are as follows [34]:

(1) Cracked or shaved ice which is simple and has the advantage of not freezing tissues. However, it is cumbersome and sloppy to handle and is unsuited to prolonged treatments.

(2) Use of ice in a pail for immersion of local parts.

(3) Special boxes for holding ice with padded or curtained openings for the limb.

(4) Bare ice bags and cloth bags for iced wet dressings for prolonged treatments and convenience.

(5) A double chambered cabinet using dry ice has been constructed.

(6) Electrical refrigerating apparatus, consisting of a compact noiseless unit that pumps fluid to various types of applications, is available. The applicators may be in the form of blankets containing rubber tubes suitable for covering the entire body or all or part of a limb. Special applicators are available for insertion into various body cavities and for inducing dental anesthesia.

(7) An air chamber at regulated temperature for treatments of frostbite and immersion foot, and amputation stumps.

The electrical apparatus is costly but has the advantages of thermostatic regulation, light weight, freedom of movement, and permits prolonged treatments with heat as well as cold over the range of temperatures therapeutically desirable.

ALLERGIC DISORDERS

Although there is some division of opinion over the ultimate cause of allergy, the prevailing belief is that it is due to an inherited or acquired hypersensitiveness to pollen or other foreign proteins in certain individuals who react abnormally to the offending substance. The reaction may be induced by inhalation, eating, or absorption (through the skin) of the allergens. Some of the clinical manifestations are hay fever, asthma, eczema, and contact dermatitis.

Symptoms of Hay Fever and Asthma

The respiratory tract is the usual site of allergic manifestations, *e.g.* hay fever and asthma. In hay fever, the nose and eyes are red and itchy, and there is considerable discharge. Nasal obstruction is the most common and distressing symptom. The severity of the symptoms varies widely from day to day depending chiefly on the amount of pollen in the air.

Seasonal asthma comes in attacks. The most popular theory concerning the mechanism of action is that the offending substance irritates the nerve endings in mucous membranes of the respiratory tract, causing spasmodic contraction of the small bronchioles of the lungs, which interferes with breathing, particularly with expiration. Non-seasonal allergic disturbances are sometimes attributed to house or street dusts, fungi, odors, animal dander, irritating gases, and heat or cold, particularly sudden temperature changes. It is often stated in the literature that heat regulation in asthmatic individuals tends to be unstable, with a tendency toward the subnormal. Many allergic cases who are apparently well develop their attacks when cold weather appears, or upon changing from warm to cool outdoor air.

Air Conditioning Apparatus

In recent years considerable effort has been directed toward the elimination of the principal cause of allergy from the air of enclosures by filtration or other air conditioning processes capable of removing pollens, in the hope of providing relief to individuals who fail to respond to medical treatment (desensitization or immunization).

Paper or cloth filters, mounted in inexpensive window or floor units, prove quite satisfactory in many cases, but since dust and smoke frequently cause asthmatic attacks, it is desirable that an air filter, to be of full value in the treatment of asthma, should remove all possible dusts

and pollens regardless of size or amount. Electrostatic air cleaners are more efficient than most commonly used types for capturing very fine dust [35].

Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, a certain amount of cooling and dehumidification appears to be desirable. A comfortable temperature between 70 and 75 F and a relative humidity well below 50 per cent proved satisfactory [36]. Direct drafts, overcooling or overheating are apt to initiate or aggravate the symptoms.

Limitations of Air Conditioning Methods

The results obtained with air filtration or other air conditioning processes in the control of allergic conditions are fairly comparable to those obtained by desensitization treatment so long as the patients remain in the pollen free atmosphere. But while specific desensitization is preventive and in a few instances curative, for all practical purposes filtration gives only temporary relief. In mild cases sleeping in an air conditioned space may make it possible for the individual to pass more comfortable days. With rare exceptions, the symptoms recur on exposure to pollen laden air. Moreover the usefulness of air conditioning methods is limited because all cases are not caused by air-borne substances. Cases of bacterial asthma do not respond to treatment with filtered air.

Despite these limitations air conditioning methods possess definite advantages in the simplicity of treatment, convenience, and under certain conditions almost immediate relief [37]. Pollen cases are usually relieved of most of their symptoms within 1 to 3 hr after exposure to properly filtered air. A pollen-free atmosphere is especially valuable when desensitization has given little or no relief, and when desensitization is not advisable owing to intercurrent illness.

OXYGEN THERAPY

Oxygen therapy is the principal measure employed for preventing and relieving the distressing symptoms of anoxemia, which is a deficiency in the oxygen content of the blood. Some of the more important conditions in which oxygen treatment is believed to be beneficial are pneumonias, anemia, heart affections, post-operative pulmonary disturbances, certain mental disturbances, asphyxia, asthma and atelectasis in new-born infants.

The effectiveness of oxygen therapy depends upon the manner of administration. Three common methods, catheter, face mask or tent [38, 39] are employed.

The necessity of air conditioning in oxygen therapy arises from the fact that oxygen is too expensive a gas to waste in the ventilation of oxygen tents and oxygen chambers. The oxygen rich atmosphere in these enclosures is therefore reconditioned in a closed circuit by removal of excess heat, moisture, and carbon dioxide given off from the occupants being treated.

Oxygen Tents

In oxygen tents the air enriched with oxygen is usually circulated by means of a small motor blower which sends the air over soda lime to remove carbon dioxide and then over ice to remove excess heat and moisture. The concentration of oxygen in the tent is regulated by means

of a pressure reducing valve and flow meter. In an inadequately cooled tent, high temperatures and humidities are inevitable, increasing the discomfort of the patient and imposing an added strain on an already overburdened heart. Oxygen therapy under such conditions may do more harm than good. An ice melting rate of approximately 10 lb per hour gives satisfactory results in patients with fever in a medium size oxygen tent.

Oxygen tents are confining to the patient. They may terrify the restless and delirious patient. Medical and nursing care is complicated, as the tent must be opened or removed with attendant loss of oxygen. Oxygen concentrations of 50 per cent or more are difficult to maintain, and it is a problem to keep the temperature and humidity low enough in hot weather. However, with attention to details, the patient can be made quite comfortable.

Oxygen Chambers

The conventional oxygen chamber is an air-tight sheet metal enclosure of fire-proof construction, large enough to accommodate one or two patients. Trap doors or curtains are provided for the personnel, food and service, to avoid loss of oxygen. Glass windows in the ceiling and walls admit light from outside the chamber. The air conditioning system may be of the gravity type, or of the fan type using mechanical refrigeration or air drying agents.

The temperature and humidity requirement in oxygen therapy depends primarily upon the physical condition of the patient, and secondarily upon the type of disease. In pneumonias[40] prescribed conditions should be a temperature of 60 to 75 F, humidity 20-50 per cent, moderate air movement, oxygen concentration of 50 per cent, and carbon dioxide of less than one per cent.

Oxygen in Aviation

An important application of the principle of oxygen therapy is in aviation. At the present time all high altitude military airplanes in this country are provided with gaseous oxygen equipment and military personnel are required to utilize oxygen at all times while in flight above 15,000 ft, or between 12,000 to 15,000 ft for longer than two hours, or between 10,000 to 12,000 ft for longer than six hours. The use of oxygen in commercial aviation will depend on the height and duration of the flights as well as the state of health of the passengers. The necessity for portable, comfortable equipment, the possible fire hazards due to smoking, and the use of oxygen on sleeper planes are some of the difficulties facing civil airline operators. The pressure cabin airplane is a solution to the problem.

GENERAL HOSPITAL AIR CONDITIONING

Complete conditioning of a large hospital involves a capital investment and running expenses which may not be justified. In clean and quiet districts, the requirements of almost all general and private wards during the cool season of the year can be satisfactorily fulfilled by the use of conventional heating equipment in conjunction with window air supply and gravity or mechanical exhaust. Insulation against heat and sound is much more important than humidification in winter; it will also help in keeping the building cool in warm weather. Excessive outside noise and dust may require the use of silencers and air filters in the openings.

Cooling and dehumidification in warm weather are important. In new hospitals particularly, the desirability of cooling certain sections of the building should be given serious consideration. Financial reasons may preclude the cooling of the entire building, but the needs of the average hospital can be met by the use of built-in room coolers and a few portable units which can be wheeled from ward to ward when needed.

In the North and certain sections of the Pacific Coast, cooling is needed but a few days during summer, while in the South, it can be used to advantage from May to October, and in tropical climates almost continuously throughout the year.

Aside from comfort and recuperative power of the patients, cooling is of great assistance in the treatment of fevers in the new-born and in post-operative cases, in enteric disorders, fevers, heat stroke, heart failure, and in a variety of other ailments which often accompany summer heat waves.

Considerable research is in progress on the influence of air conditioning upon a wide variety of diseases such as pneumonia, upper respiratory diseases, tuberculosis, arthritis, nervous instability, hyper-thyroidism, essential hypertension, skin diseases, and vascular disorders.

Problem of Odors

The evacuation of battle casualties in aircraft and their subsequent hospitalization have stimulated efforts to minimize odors arising from draining wounds, old odorous casts, and gangrenous wounds. For aircraft, chemical sprays and vapors, perfumes, oxidizing gases and ventilation methods are unsatisfactory. An ideal deodorant would purify the air by means of odor adsorption so that subsequently the air can be recirculated. Based upon the effectiveness of activated carbon commercially and industrially to adsorb odors, individual adsorption units have been used successfully. In hospital wards the question of superiority of adsorption methods for elimination of odors over other methods remains to be answered. The present status of the problem is that the commercial aspect is highly controversial.

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CHAPTER 14

Heating Load

General Procedure, Outside Temperatures, Inside Temperatures, Attic Temperatures, Temperatures in Unheated Spaces, Ground Temperatures, Basement Temperatures and Heat Loss, Transmission Heat Loss, Heat Loss Through Ceilings and Roofs, Infiltration Loss, Selection of Wind Velocities, Auxiliary Heat Sources, Intermittently Heated Buildings, Residence Heat Loss Problems

IN the design of a heating system, an estimate must be made of the maximum probable heat loss of each room or space to be heated, based on maintaining a specified inside air temperature during periods of minimum selected design weather conditions. The heat losses may be divided into two groups, namely (1) the transmission losses or heat losses through the confining walls, floor, ceiling, glass or other surfaces and (2) the infiltration losses or heat losses due to air leakage through cracks and crevices, around doors and windows, opening of doors and other sources of interchange of air between the inside and outside.

GENERAL PROCEDURE

The general procedure for calculating heat losses of a structure is:

1. Select the outside design temperature. The data on climatic conditions given in Table 1 and the bands of average design temperature in Fig. 1 will be useful but should be applied with judgment as suggested in the section Outside Design Temperatures.

2. Select the inside air temperature, at the 60-in. breathing line or the 30-in. line which is to be maintained in the building during the coldest weather. (See Table 2).

3. Estimate temperatures in adjacent unheated spaces and the attic. The attic temperature need not be estimated if the combined roof and ceiling coefficient is used.

4. Select or compute the heat transmission coefficients for outside walls and glass; also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 6).

5. Measure amount of net outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building, using inside dimensions.

6. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet and the temperature difference between the inside and outside air. (See Items 1, 2, and 3).

7. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack and wind velocity, and when multiplied by the length of crack and the temperature difference between the inside and outside air, the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 8).

8. The sum of the heat losses by transmission (Item 6) through the outside wall and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent (Item 7) of the cold air entering by infiltration, or required to replace mechanical exhaust, represents the total heat loss equivalent for any building.

OUTSIDE DESIGN TEMPERATURES

There are no hard and fast rules for selecting the outside design temperature to be used for a given locality or type of building or heating system, and the problem is to some extent a matter of judgment and experience. The outside design temperature is seldom taken as the lowest temperature, or even the lowest daily mean temperature ever recorded in a given locality. Such temperatures are rarely repeated in successive years. A temperature somewhat higher than the minimum or the lowest daily mean on

TABLE 1. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F
State	City	Elevation[b]	Average Temperature, Oct. 1-May 1	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour
Alabama	Anniston	724	52.7	−10	
	Birmingham	694	52.9	−10	8.0
	Mobile	10	59.2	− 1	9.9
	Montgomery	201	56.7	− 5	7.5
Arizona	Flagstaff	6902	36.0	−30	
	Phoenix	1112	46.3	16	5.3
	Yuma	138	62.5	22	6.7
Arkansas	Bentonville	1295	46.9	−17	
	Fort Smith	449	50.8	−15	8.1
	Little Rock	257	51.8	−12	8.3
California	Eureka	43	49.3	20	7.3
	Fresno	277	54.3	17	5.4
	Independence	3944	47.8	− 5	
	Los Angeles	312	60.0	28	6.3
	Needles	480	60.8	18	
	Point Reyes	510	51 7	27	
	Red Bluff	303	53.1	17	6.0
	Sacramento	25	53.3	17	7.2
	San Diego	19	57.9	25	6.3
	San Francisco	52	54.6	27	7.5
	San Jose	95	53.3	18	
Colorado	Denver	5221	39.7	−29	7.6
	Durango	6552	35.4	−27	
	Grand Junction	4587	39.7	−21	4.4
	Leadville	10182	24.9	−31	
	Pueblo	4799	40.3	−27	8.0
Connecticut	Hartford	15	36.4	−24	8.9
	New Haven	13	38.6	−15	9.4
Dist. of Columbia	Washington	72	43.4	−15	7.9
Florida	Apalachicola	14	61.2	18	8.4
	Jacksonville	18	62.7	10	9.1
	Key West	6	73.1	43	10.6
	Miami	8	71.2	27	9.8
	Pensacola	13	60.0	7	11.2
	Tampa	24	66.0	19	8.5
Georgia	Atlanta	987	52.1	− 8	11.7
	Augusta	134	55.3	3	6.5
	Macon	330	55.0	7	6.6
	Savannah	38	56.9	8	9.6
	Thomasville	276	58.9	2	5.3
Idaho	Boise	2842	40.3	−13	9.1
	Lewiston	738	44.7	−23	4.1
	Pocatello	4468	36.5	−28	9.5
Illinois	Cairo	314	46.9	−16	9.9
	Chicago	594	37 3	−23	12.0
	Peoria	603	37.4	−27	8.1
	Springfield	602	40.5	−24	11.8
Indiana	Evansville	385	45.1	−16	9.7
	Fort Wayne	777	36.7	−24	10.3
	Indianapolis	718	40.4	−25	11.3
	Royal Center	736	37.8	−25	11.0
	Terre Haute	485	42.0	−18	10.2
Iowa	Charles City	1013	31.0	−34	7.9
	Davenport	579	36.4	−27	10.5
	Des Moines	800	36.0	−30	10.1
	Dubuque	641	34.2	−32	7.1
	Keokuk	574	38.9	−27	8.3
	Sioux City	1093	33.9	−35	11.6
Kansas	Concordia	1375	40.7	−25	7.7
	Dodge City	2522	42.3	−26	10.5

Table 1. Climatic Conditions Compiled from Weather Bureau Records[a]—(Continued)

Col. A	Col. B	Col. C	Col. D	Col. E	Col. F
State	City	Elevation[b]	Average Temperature, Oct. 1-May 1	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour[c]
Kansas	Iola	963	44.8	−18	8.2
	Topeka	926	42.1	−25	9.6
	Wichita	1372	44.1	−22	12.2
Kentucky	Louisville	539	44.8	−20	9.9
	Lexington	975	43.7	−20	13.3
Louisiana	New Orleans	3	62.1	7	8.6
	Shreveport	174	56.3	− 5	8.8
Maine	Eastport	33	31.5	−23	12.5
	Greenville	1057	25.9	−36	
	Portland	61	34.0	−21	10.4
Maryland	Baltimore	14	44.2	− 7	8.1
Massachusetts	Boston	12	38.3	−18	12.4
	Fitchburg	402	35.3	−17	
	Nantucket	43	39.3	− 6	14.8
Michigan	Alpena	587	29.9	−28	11.0
	Detroit	619	35.8	−24	12.0
	Escanaba	594	27.8	−32	9.3
	Grand Rapids	638	35.2	−24	11.9
	Houghton	695	27.4	−31	9.1
	Lansing	858	34.0	−25	10.0
	Ludington	690	33.8	−21	
	Marquette	674	31.2	−27	10.7
	Sault Ste. Marie	607	26.7	−37	8.9
Minnesota	Duluth	1128	24.3	−41	13.4
	Minneapolis	830	29.8	−34	11.3
	Moorhead	895	22.8	−48	9.9
	St. Paul	703	29.0	−34	9.5
Mississippi	Corinth	470	51.0	− 8	
	Meridian	343	54.8	− 6	5.7
	Vicksburg	234	56.8	− 1	8.3
Missouri	Columbia	733	42.3	−26	8.9
	Hannibal	759	40.1	−25	9.6
	Kansas City	741	42.8	−22	10.2
	St. Louis	465	44.3	−22	11.7
	Springfield	1265	44.8	−29	10.9
Montana	Billings	3568	33.4	−38	12.4
	Havre	2488	27.6	−57	9.4
	Helena	3893	29.1	−42	7.3
	Kalispell	2956	31.6	−34	5 2
	Miles City	2351	27.6	−49	5.6
Nebraska	Drexel	1299	35.7	−25	
	Lincoln	1184	37.8	−29	10.6
	North Platte	2783	36.8	−35	8.3
	Omaha	978	35.8	−32	9.6
	Valentine	2581	33.5	−38	9.3
Nevada	Reno	4397	39.0	−19	6.0
	Tonopah	6087	39.7	−15	
	Winnemucca	4288	37.9	−36	8.1
New Hampshire	Concord	339	31.0	−37	6.0
New Jersey	Atlantic City	8	41.6	− 9	15.9
	Cape May	17	43.2	− 3	
	Newark	11	40.3	−14	
	Sandy Hook	15	41.2	−11	16.1
	Trenton	56	40.6	−14	11.0
New Mexico	Albuquerque	5314	44.1	−16	7.3
	Roswell	3568	48.9	−29	7.1
	Santa Fe	6312	38.6	−13	7.1
New York	Albany	119	35.4	−24	10.4
	Binghamton	858	35.1	−28	6.8

Table 1. Climatic Conditions Compiled from Weather Bureau Records[a]—(Continued)

Col. A	Col. B	Col. C	Col. D	Col. E	Col. F
State	City	Elevation[b]	Average Temperature, Oct. 1-May 1	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour
New York	Buffalo	693	35.0	−20	17.1
	Canton	406	29.5	−43	10.6
	Ithaca	872	35.1	−22	11.3
	New York	10	41.0	−14	16.8
	Oswego	292	34.2	−23	12.3
	Rochester	543	35.1	−22	10.1
	Syracuse	399	35.0	−26	11.3
North Carolina	Asheville	2203	45.9	− 6	9.5
	Charlotte	753	50.5	− 5	7.3
	Hatteras	7	53.4	8	14.6
	Manteo	12	51.6	7	
	Raleigh	400	50.1	− 2	7.8
	Wilmington	6	54.3	5	8.1
North Dakota	Bismarck	1650	25.3	−45	9.1
	Devils Lake	1472	21.8	−46	10.1
	Grand Forks	832	22.2	−43	
	Williston	1877	23.4	−50	8.4
Ohio	Cincinnati	761	42.3	−17	8.5
	Cleveland	787	37.6	−17	15.0
	Columbus	724	40.2	−20	11.6
	Dayton	743	40.3	−28	10.9
	Sandusky	603	38.0	−16	11.0
	Toledo	589	37.2	−16	12.1
Oklahoma	Broken Arrow	765	47.9	−12	
	Oklahoma City	1280	48.7	−17	11.5
Oregon	Baker	3445	35.9	−25	5.9
	Medford	1314	45.1	−10	4.3
	Portland	30	47.2	− 2	7.3
	Roseburg	479	47.1	− 6	3.9
Pennsylvania	Erie	654	37.0	−16	13.6
	Harrisburg	335	40.6	−14	7.9
	Philadelphia	26	43.5	−11	11.0
	Pittsburgh	1248	40.7	−20	11.7
	Reading	266	40.9	−14	9.1
	Scranton	746	37.5	−19	7.6
Rhode Island	Block Island	35	40.0	−10	18.0
	Narragansett Pier	22	38.1	−13	
	Providence	12	38.7	−17	12.1
South Carolina	Charleston	9	54.6	7	10.5
	Columbia	332	54.2	− 2	8.1
	Due West	711	51.6	6	8.6
	Greenville	1006	49.1	− 5	8.4
South Dakota	Huron	1282	29.2	−43	10.6
	Pierre	1719	32.4	−40	8.3
	Rapid City	3215	34.0	−34	7.8
Tennessee	Chattanooga	689	50.2	−10	7.6
	Knoxville	949	48.1	−16	7.2
	Memphis	269	51.2	− 9	9.3
	Nashville	585	48.9	−13	9.8
Texas	Abilene	1752	54.2	− 6	10.1
	Amarillo	3590	43.3	−16	12.1
	Austin	615	57.0	− 1	8.0
	Brownsville	16	66.8	12	10.2
	Corpus Christi	10	61.9	11	11.0
	Dallas	487	55.4	− 3	10.6
	Del Rio	957	60.7	12	7.9
	El Paso	3710	53.8	− 5	8.9
	Ft. Worth	688	55.6	− 8	10.4
	Galveston	7	62.0	8	11.2

TABLE 1. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]—(Concluded)

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F
State	City	Elevation[b]	Average Temperature, Oct. 1-May 1	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour
Texas	Houston	41	61.6	5	10.5
	Palestine	492	57.0	− 6	8.0
	Port Arthur	5	61.1	11	10.5
	San Antonio	782	60.9	4	8.3
	Taylor	583	58.2	0	9.2
Utah	Modena	5460	36.7	−32	9.3
	Salt Lake City	4222	38.6	−30	7.9
Vermont	Burlington	398	31.2	−29	11.7
	Northfield	842	28.1	−41	
Virginia	Cape Henry	16	49.0	5	13.1
	Lynchburg	631	46.4	− 7	8.1
	Norfolk	11	49.6	2	10.1
	Richmond	162	47.4	− 3	8.1
	Wytheville	2299	42.0	− 8	
Washington	North Head	194	43.3	11	16.3
	Seattle	14	45.9	3	9.7
	Spokane	1955	37.9	−30	6.2
	Tacoma	109	44.9	7	7.9
	Tatoosh Island	101	44.4	7	19.9
	Walla Walla	949	43.4	−29	5.5
	Yakima	1068	40.6	−24	4.1
West Virginia	Elkins	1969	38.2	−28	6.1
	Parkersburg	615	42.8	−27	7.1
Wisconsin	Green Bay	593	30.4	−36	10.6
	La Crosse	664	31.5	−43	8.9
	Madison	938	31.8	−29	10.1
	Milwaukee	674	33.6	−25	12.1
	Wausau	1221	26.1	−40	
Wyoming	Cheyenne	6139	33.9	−38	13.3
	Lander	5352	30.1	−40	3.9
	Yellowstone Park	6239	29.4	−40	8.8
Alta	Edmonton	2219	22.8	−57	7.5
B. C.	Vancouver	22	42.6	2	4.5
	Victoria	228	44.0	− 2	12.6
Man	Winnipeg	786	17.2	−54	10.1
N. B.	Fredericton	164	27.5	−35	9.1
N. S.	Yarmouth	136	34.8	−12	14.3
Ont	London	912	32.6	−27	10.3
	Ottawa	294	26.4	−35	8.4
	Port Arthur	644	22.0	−40	8.0
	Toronto	379	32.6	−26	13.6
P. E. I.	Charlottetown	186	29.4	−27	9.8
Que	Montreal	187	28.1	−29	11.3
	Quebec	296	24.5	−34	13.3
Sask	Prince Albert	1414	16 0	−70	5.1
Yukon	Dawson	1062	1.9	−68	3.7
Newfoundland	St. Johns	428	31.4	−21	12

[a]United States data from U S Weather Bureau, and Canadian data from Meteorological Service of Canada, corrected to 1943.
[b]Elevation of ground at station above sea level, feet. Elevations obtained—1946.
[c]Blank spaces indicate data not available.

record may properly be assumed in making the heat loss computations. Column E, Table 1 lists the lowest dry-bulb temperatures ever reported in the places listed. Temperatures for other cities may be obtained from local weather bureau records[1]. The design temperatures shown in Fig. 1 are generally representative of the practice in various sections of the United States, although in some instances, due to local con-

FIG. 1. BANDS OF AVERAGE WINTER DESIGN TEMPERATURE

ditions of altitude or exposure, the design temperatures may vary somewhat from those indicated.

The A.S.H.V.E. Technical Advisory Committee on Weather Design Conditions has recommended the adoption for heating of an outside design temperature which is equalled or exceeded during 97½ per cent

of the hours in December, January, February, and March, but the work of compiling the various local temperatures is not yet completed.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures, usually specified, vary in accordance with the use to which the building is to be put and Table 2 presents values which conform to good practice.

TABLE 2. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED[a]

TYPE OF BUILDING	DEG F	TYPE OF BUILDING	DEG F
SCHOOLS—		THEATERS—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70		
Wardrobe and locker rooms	65–68	HOTELS—	
Kitchens	66	Bedrooms and baths	70
Dining and lunch rooms	65–70	Dining rooms	70
Playrooms	60–65	Kitchens and laundries	66
Natatoriums	75	Ballrooms	65–68
		Toilets and service rooms	68
HOSPITALS—			
Private rooms	70–72	HOMES	70–72
Private rooms (surgical)	70–80	STORES	65–68
Operating rooms	70–95	PUBLIC BUILDINGS	68–72
Wards	68	WARM AIR BATHS	120
Kitchens and laundries	66	STEAM BATHS	110
Toilets	68	FACTORIES AND MACHINE SHOPS	60–65
Bathrooms	70–80	FOUNDRIES AND BOILER SHOPS	50–60
		PAINT SHOPS	80

[a]The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion. These three factors considered together constitute what is termed the *effective temperature*. (See Chapter 12.) When relative humidity is not controlled separately, optimum dry-bulb temperature for comfort will be slightly higher than shown in Table 2.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 12. In other words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter *effective temperature* for sedentary persons, as determined at the A.S.H.V.E. Research Laboratory, is 66 deg.

As explained in Chapter 12 for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective temperature of 66 deg when the dry-bulb temperature is 70 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures listed in Table 2, consideration should be given to the actual relative humidity to be maintained if provision is to be made for humidification.

Temperature at Proper Level: In making the actual heat loss computations, however, for the various rooms in a building it is often necessary

to modify the temperatures given in Table 2 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level.

Temperature at Ceiling: The air temperature at the ceiling is generally higher than at the breathing level due to stratification of air resulting from the tendency of the warmer or less dense air to rise. An allowance for this fact should be made in calculating ceiling heat losses, particularly in the case of high ceilings. However, the exact allowance to be made may be

TABLE 3. APPROXIMATE TEMPERATURE DIFFERENTIALS BETWEEN BREATHING LEVEL AND CEILING, APPLICABLE TO CERTAIN TYPES OF HEATING SYSTEMS[a]

CEILING HEIGHT (FT)	BREATHING LEVEL TEMPERATURE (5 FT ABOVE FLOOR)									
	60	65	70	72	74	76	78	80	85	90
10	3.0	3.3	3.5	3.6	3.7	3.8	3.9	4.0	4.3	4.5
11	3.6	3.9	4.2	4.3	4.4	4.6	4.7	4.8	5.1	5.4
12	4.2	4.6	4.9	5.0	5.2	5.3	5.5	5.6	6.0	6.3
13	4.8	5.2	5.6	5.8	5.9	6.1	6.2	6.4	6.8	7.2
14	5.4	5.9	6.3	6.5	6.7	6.8	7.0	7.2	7.7	8.1
15	6.0	6.5	7.0	7.2	7.4	7.6	7.8	8.0	8.5	9.0
16	6.1	6.6	7.1	7.3	7.5	7.7	7.9	8.1	8.6	9.1
17	6.2	6.7	7.2	7.4	7.6	7.8	8.0	8.2	8.7	9.2
18	6.3	6.8	7.3	7.5	7.7	7.9	8.1	8.3	8.8	9.3
19	6.4	6.9	7.4	7.6	7.8	8.0	8.2	8.4	8.9	9.4
20	6.5	7.0	7.5	7.7	7.9	8.1	8.3	8.5	9 0	9.5
25	7.0	7.5	8.0	8.2	8.4	8.6	8.8	9.0	9.5	10.0
30	7.5	8.0	8.5	8.7	8.9	9.1	9.3	9.5	10.0	10.5
35	8.0	8.5	9.0	9.2	9.4	9.6	9.8	10.0	10.5	11.0
40	8.5	9.0	9.5	9.7	9.9	10.1	10.3	10.5	11.0	11.5
45	9.0	9.5	10.0	10.2	10.4	10.6	10.8	11.0	11.5	12.0
50	9.5	10.0	10.5	10.7	10.9	11.1	11.3	11.5	12.0	12.5

[a]The figures in this table are based on an increase of 1 per cent per foot of height above the breathing level (5 ft) up to 15 ft and 1/10 of one degree for each foot above 15 ft. This table is generally applicable to forced air types of heating systems. For direct radiation or gravity warm air, increase values 50 per cent to 100 per cent.

somewhat difficult to determine as it depends on many factors, including (1) the type of heating system, (2) ceiling height, and (3) the inside-outside temperature differential. The type of heating system is particularly important as the temperature gradient from floor to breathing-level to ceiling may depend to a large extent on whether direct radiation, unit heaters or warm air is used, and in the latter case, whether the circulation is by gravity, auxiliary fan or forced air. Although with properly adjusted air flow the temperature differential with unit heaters can be reduced to a minimum, it is possible with improper adjustment that it may be increased over that which would normally result without mechanical circulation of the air.

It would be difficult from present available information to establish rules for determining the temperature difference to use in all cases. However, for residences and other structures having ceiling heights under 10 ft, the comparatively small temperature differential between the

breathing level and ceiling may generally be neglected without serious error. For higher ceilings where specific test data are not available, an allowance of approximately 1 per cent per foot of height above the breathing level may be made for ceiling heights up to 15 ft and approximately 1/10 of 1 deg per foot of height above this level. The values in Table 3 are calculated on this basis. For direct radiation and gravity warm air systems, the allowance should be increased from 50 per cent to 100 per cent over those given in Table 3. These rules should, however, be used with considerable discretion.

Temperature at Floor Level: According to the University of Illinois Research Residence tests [2], the temperature at the floor level ranged from about 2½ to 6 deg below that at the breathing level, or somewhat greater than the difference between the breathing level and ceiling temperatures. Tests at the University of Wisconsin [3] indicated a somewhat smaller differential between the floor and breathing level temperatures. As a general rule, if the breathing level to ceiling temperature differential is neglected (as with ceiling heights under 10 ft), the breathing level-floor differential may also be neglected as the two are somewhat compensating, especially where both floor and ceiling heat losses are calculated for the same space. In other cases, the 10 ft temperature differentials in Table 3 may be used in arriving at the floor heat loss, these differentials to be subtracted from the breathing level temperature.

ATTIC TEMPERATURES

Frequently it is necessary to estimate the attic temperature, and in such cases Equation 1 can be used for this purpose.

$$t_a = \frac{A_c U_c t_i + t_o (A_r U_r + A_w U_w + A_g U_g)}{A_r U_r + A_w U_w + A_g U_g + A_c U_c} \qquad (1)$$

where

t_a = attic temperature, Fahrenheit degrees.
t_i = inside temperature near top floor ceiling, Fahrenheit degrees.
t_o = outside temperature, Fahrenheit degrees.
A_c = area of ceiling, square feet.
A_r = area of roof, square feet.
A_w = area of net vertical attic wall surface, square feet.
A_g = area of attic glass, square feet.
U_c = coefficient of transmission of ceiling, based on surface conductance of 2.20 (upper surface, see Chapter 6). 2 20 = reciprocal of one-half the air space resistance.
U_r = coefficient of transmission of roof, based on surface conductance of 2.20 (lower surface, see Chapter 6).
U_w = coefficient of transmission of vertical wall surface.
U_g = coefficient of transmission of glass.

Example 1. Calculate the temperature in an unheated attic, assuming the following conditions: $t_i = 70$; $t_o = 10$; $A_c = 1000$; $A_r = 1200$; $A_w = 100$; $A_g = 10$; $U_r = 0.50$; $U_c = 0.40$; $U_w = 0.30$; $U_g = 1.13$.

Solution: Substituting these values in Equation 1:

$$t_a = \frac{(1000 \times 0.40 \times 70) + 10\,[(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)]}{(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13) + (1000 \times 0.40)}$$

$$t_a = \frac{34{,}413}{1041} = 33.1 \text{ F.}$$

Equation 1 neglects the effect of any interchange of air such as would take place through attic vents or louvers intended to preclude attic.condensation. However, according to tests [4], such venting of attics by means of small louvers or other small openings does not appreciably reduce the attic temperature and may be neglected without serious error. The attic temperature may be calculated in the usual manner by means of Equation 1, allowing the full value of the roof. The error resulting from this assumption will generally be considerably less than if the roof were neglected (as is sometimes the practice) and the attic temperature assumed to be the same as the outside temperature. When relatively large louvers are installed as is customary in the southern states, the attic temperature is often assumed as the average between inside and outside temperatures.

TEMPERATURES IN UNHEATED SPACES

The heat loss from heated rooms into unheated rooms or spaces must be based on the estimated or assumed temperature in such unheated spaces. This temperature will generally range between the inside and outside temperatures, depending on the relative areas of the surfaces adjacent to the heated room and exposed to the outside. If the respective surface areas adjacent to the heated room and exposed to the outside are approximately the same, and if the coefficients of transmission are approximately equal, the temperature in the unheated space may be assumed to be the mean of the inside and outside design temperatures. If, however, the surface areas and coefficients are unequal, the temperature in the unheated space should be estimated by means of Equation 2.

$$t_u = \frac{t(A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.}) + t_o\,(A_aU_a + A_bU_b + A_cU_c + \text{etc.})}{A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.} + A_aU_a + A_bU_b + A_cU_c + \text{etc}} \qquad (2)$$

where

t_u = temperature in unheated space, Fahrenheit degrees.

t = inside design temperature of heated room, Fahrenheit degrees.

t_o = outside design temperature, Fahrenheit degrees.

A_1, A_2, A_3, etc. = areas of surface of unheated space adjacent to heated space, square feet.

A_a, A_b, A_c, etc. = areas of surface of unheated space exposed to outside, square feet.

U_1, U_2, U_3, etc. = coefficients of transmission of surfaces of A_1, A_2, A_3, etc.

U_a, U_b, U_c, etc. = coefficients of transmission of surfaces A_a, A_b, A_c, etc.

Example 2. Calculate the temperature in an unheated space adjacent to a heated room having surface areas (A_1, A_2, and A_3) in contact therewith of 100, 120, and 140 sq ft and coefficients (U_1, U_2, and U_3) of 0 15, 0.20, and 0.25 respectively. The surface areas of the unheated space exposed to the outside (A_a and A_b) are respectively 100 and 140 sq ft and the corresponding coefficients are 0.10 and 0.30. The sixth surface is on the ground and is neglected in this example. Assume $t = 70$ and $t_o = -10$.

Solution. Substituting in Equation 2:

$$t_u = \frac{70[(100\times0.15)+(120\times0.20)+(140\times0.25)]+-10[(100\times0.10)+(140\times0.30)]}{(100\times0.15)+(120\times0.20)+(140\times0.25)+(100\times0.10)+(140\times0.30)}$$

$$t_u = \frac{4660}{126} = 37\text{ F.}$$

The temperatures in unheated spaces having large glass areas and with two or more surfaces exposed to the outside (such as sleeping porches and sun parlors), are generally assumed to be the same as outside.

GROUND TEMPERATURES

Ground temperatures to be assumed for estimating basement heat losses will usually differ in the case of basement walls and floors, the temperatures under the floors being generally higher than those adjacent to walls.

Temperatures Adjacent to Basement Walls

Ground temperatures near the surface and under open spaces vary with the climate, the season of the year and the depth below the surface. The nearer the surface (during the cold weather) the lower the temperature. Frost will penetrate to a depth of over 4 ft in some localities if not protected by snow. A thick blanket of snow will result in a higher ground temperature near the surface. Consequently ground temperatures near the surface may be higher in cold climates where the snow remains on the ground for a greater length of time than in more moderate climates where the snow melts away periodically during the winter.

Complete data for various localities are not as yet available but in estimating heat losses through vertical walls below grade, it is advisable not to assume average ground temperatures above 32 F in northern climates when estimating heat losses from heated basements. This is for the mean height of the basement wall. Since the recommended wall coefficient for basement walls in contact with the soil is only 0.10, any small variation in the assumed ground temperature will not materially affect the calculated heat loss.

Temperatures Under Basement Floors

The temperature under basement floors [5] is influenced by the heat from the basement or protected from the influence of atmospheric conditions by the basement. In computing losses through basement floors the ground temperatures may be assumed the same as the approximate water temperature at depths of 30 to 60 ft given in Fig. 3, Chapter 37. Test results indicate that the heat losses through basement floors are frequently over-estimated [6].

BASEMENT TEMPERATURES AND HEAT LOSS

The allowance to be made for basement heat loss depends on whether the basement is to be heated or not.

If the basement is *heated* and a specified temperature is to be maintained, the heat loss should be estimated in the usual manner, based on the proper wall and floor coefficients (see Chapter 6) and the outside air and ground temperatures. Heat loss through windows and walls above grade should be based on outside temperatures and the proper air-to-air coefficients. Heat loss through basement walls below grade should be based on the floor and wall coefficients for surfaces in contact with the soil and on the proper ground temperature.

If a basement is completely below grade and is *not heated,* the temperature in the basement will normally range between that in the rooms

above and the ground temperature. Basement windows will of course lower the basement temperature when it is colder outside and any heat given off by the heating plant will increase the basement temperature. In any case, the exact basement temperature is likely to be a somewhat indeterminate quantity, if the basement is not heated. Since the basement temperature will generally be lower than that of the rooms above, an allowance should theoretically be made for the loss from the rooms above through the floor over the basement.

The temperature in crawl spaces below floors will vary greatly depending on the number and size of wall vents, the quantity of heating pipes and the type of insulation. It is therefore necessary to analyze the conditions and select an appropriate temperature by judgment.

TRANSMISSION HEAT LOSS

The basic formula for the loss of heat by transmission through any surface is given in Equation 3.

$$H_t = A\,U\,(t - t_o) \tag{3}$$

where

H_t = heat loss transmitted through the wall, roof, ceiling, floor, or glass, Btu per hour.

A = area of wall, glass, roof, ceiling, floor, or other exposed surfaces, square feet.

U = coefficient of transmission, air to air, Btu per (hour) (square foot) (Fahrenheit degree temperature difference) (Chapter 6).

t = inside temperature near surface involved which may not necessarily be the so-called breathing line temperature, Fahrenheit degrees.

t_o = outside temperature, or temperature of adjacent unheated space or of the ground, Fahrenheit degrees.

Example 3 Calculate the transmission loss through an 8 in. brick wall having an area of 150 sq ft if the inside temperature (t) is 70 F and the outside temperature (t_o) is −10 F.

Solution. The coefficient of transmission (U) of a plain 8 in. brick wall is 0.50 (Chapter 6, Table 7). The area (A) is 150 sq ft. Substituting in Equation 3:

$$H_t = 150 \times 0.50 \times [70 - (-10)] = 6000 \text{ Btu per hour.}$$

Transmission Loss Through Ceilings and Roofs

The transmission heat loss through top floor ceilings, attics, and roofs may be estimated by either of two methods:

1. By substituting in Equation 3 the ceiling area (A), the inside-outside temperature difference ($t - t_o$) and the proper value of (U):

   a. *Flat roofs* Select the coefficient of transmission of the ceiling and roof from Tables 14 or 15, Chapter 6, or use appropriate coefficients in Equation 1 if side walls extend appreciably above the ceiling of the floor below.

   b. *Pitched roofs.* Select the combined roof and ceiling coefficient from Table 17, Chapter 6 or calculate the combined roof and ceiling coefficient by means of Equation 5, Chapter 6, where this formula is applicable as explained in Chapter 6.

2. By estimating the attic temperature (based on the inside and outside design temperatures) by means of Equation 1, and substituting for t_o in Equation 3, the value of t_a thus obtained, together with the ceiling area (A) and the ceiling coefficient (U). This applies to *pitched roofs*. In the case of *flat roofs* it is not necessary to calculate the attic temperatures as the ceiling-roof heat loss can be determined as per paragraph 1a.

INFILTRATION HEAT LOSS

The infiltration heat loss includes (1) the *sensible heat loss* or the heat required to warm the outside air entering by infiltration and (2) the *latent heat loss* or the heat equivalent of any moisture which must be added.

Sensible Heat Loss

The formula for the heat required to warm the outside air which enters a room by infiltration to the temperature of the room, is given in Equation 4.

$$H_s = 0.24\,Qd\,(t - t_o) \tag{4}$$

where

H_s = heat required to raise temperature of air leaking into building from t_o to t, Btu per hour.

0.24 = specific heat of air.

Q = volume of outside air entering building, cubic feet per hour (see Chapter 8).

d = density of air at temperature t_o, pounds per cubic foot.

It is sufficiently accurate to use $d = 0.075$ in which case Equation 4 reduces to

$$H_s = 0.018\,Q\,(t - t_o) \tag{4a}$$

The volume of outside air entering per hour (Q) depends on the wind velocity and direction, the width of crack or size of openings, the type of openings and other factors, as explained in Chapter 8. Where the crack method is used for estimating leakage, it is more convenient to express the air leakage heat loss in terms of the crack length:

$$H_s = 0.018\,Q\,L\,(t - t_o) = B\,L\,(t - t_o) \tag{4b}$$

where

B = air leakage per (hour) (foot of crack) (Chapter 8) for the wind velocity and type of windows or door crack involved multiplied by 0.018.

L = length of window or door crack to be taken into consideration, feet.

Example 4. What is the infiltration heat loss per hour through the crack of a 3 x 5 ft average, double-hung, non-weatherstripped, wood window, based on a wind velocity of 15 mph? Assume inside and outside temperatures to be 70 F and zero respectively.

Solution. According to Table 2, Chapter 8, the air leakage through a window of this type (based on $\frac{1}{16}$ in. crack and $\frac{3}{64}$ in. clearance) is 39 cu ft per foot of crack per hour. Therefore, $B = 39 \times 0.018 = 0.70$. The length of crack ($L$) is $(2 \times 5) + (3 \times 3)$, or 19 ft; $t = 70$ and $t_o = 0$. Substituting in Equation 4b,

$$H_s = 0.70 \times 19 \times (70 - 0) = 931 \text{ Btu per hour.}$$

Crack Length to be Used for Computations

The amount of crack used for computing the infiltration heat loss should not be less than half of the total crack in the outside walls of the room. For a building having no partitions, whatever wind enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building. In a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack.

The total infiltration loss of a building having partitions will not be equal to the sum of the infiltration losses of the various rooms, since at any given time infiltration will take place only on the windward side or sides and not on the leeward side. Therefore, if a building has more than one room which is divided by interior walls or partitions, it is sufficiently accurate to use half of the total infiltration losses for determining the total heat requirements. Infiltration is sometimes estimated in terms of number of air changes per hour. Such estimates require judgment regarding the construction and conditions under consideration, that is, whether they are better, equal to, or worse than the average conditions for which air changes are given in Table 4, Chapter 8.

Latent Heat Loss

When it is intended to add moisture to air leaking into a room for the maintenance of proper winter comfort conditions, it is necessary to determine the heat equivalent to evaporate the required amount of water vapor, which may be calculated by the equation:

$$H_l = Q\,d\left(\frac{m_i - m_o}{7000}\right) h_{fg} \tag{5}$$

where

H_l = heat required to increase moisture content of air leaking into building from m_o to m_i, Btu per hour.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_i, pounds per cubic foot.
m_i = vapor density of inside air, grains per pound of dry air.
m_o = vapor density of outside air, grains per pound of dry air.
h_{fg} = latent heat of vapor at m_i, Btu per pound.

If the latent heat of vapor (h_{fg}) is assumed to be 1060 Btu per pound, Equation 5 reduces to

$$H_l = 0.0114\,Q\,(m_i - m_o) \tag{5a}$$

Equations 4a, 4b and 5a may also be used for determining the sensible and latent heat gains due to infiltration in cooling load computations.

SELECTION OF WIND VELOCITIES

The effect of wind on the heating requirements of any building should be given consideration under two heads:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls.

2. Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves, if such materials are at all porous.

Theoretically as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature which is the worst would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to compute the heating load for a building for several different combinations of temperature and wind velocity which

records show to have occurred and to select the worst combination; but designers generally do not feel that such a degree of refinement is justified.

It has been the practice for many years in estimating air leakage by the crack method to use the *average* wind velocity during the months of December, January and February. This *average* wind velocity may not necessarily correspond with that occurring during periods when the outside design temperature prevails, the latter being not an *average* but rather a near extreme, that is, a specified number of degrees above the lowest temperature recorded in the locality involved. Therefore instead of using the aforementioned *average* wind velocity, it is the practice of some designers to use in all cases a wind velocity of 15 mph together with the proper design temperature. Although a 15 mph wind velocity is higher than the general average wind velocity during December, January and February in various United States cities, this and higher wind velocities frequently occur during periods of outside temperature corresponding to the design temperature. It should be added that this wind velocity also corresponds with that on which the heat loss coefficients in Chapter 6 are based, although the effect of variations in wind velocity on the infiltration losses is generally much greater than the effect of wind velocity on the heat loss by transmission through walls. Therefore, pending further investigation of this subject, either the average during December, January and February or a 15 mph wind velocity may be used at the discretion of the designer. Where the air change method is used for estimating infiltration losses, the wind velocity is not considered.

Exposure Factors

Many designers use empirical exposure factors to increase the calculated heat loss of rooms or spaces on the side or sides of the building exposed to the prevailing winds. However, according to a survey made in 1943, many GUIDE users have found that the use of exposure factors is not necessary as the GUIDE method of calculating heat losses provides an ample heat loss allowance. Therefore exposure factors may be regarded as *factors of safety* for the rooms or spaces exposed to the prevailing winds, to allow for additional capacity for these rooms or spaces, or to *balance the radiation*, particularly in the case of multi-story buildings. Although the exposure allowance is frequently assumed to be 15 per cent, the actual allowance to be made, if any, must to a large extent be a matter of experience and judgment of the designer, since there are at present no authentic test data available from which rules could be developed for the many conditions encountered in practice.

As stated previously, the value of U in the tables of Chapter 6 is based on a wind velocity of 15 mph and the surface resistance for this wind velocity (0.17) is sufficiently low so that higher wind velocities will decrease the surface resistance to a negligible degree and therefore have only a slight effect on the average over-all coefficient. On the other hand, infiltration losses vary almost directly as the wind velocity, as will be apparent from the factors in Table 2 of Chapter 8. The more exact method therefore would be to differentiate among the various exposures more accurately by calculating the infiltration and transmission losses separately for the different sides of the building, using different assumed wind velocities for the infiltration losses on the various sides of the building.

Because the prevailing wind direction does not necessarily coincide with the wind direction accompanying the coldest weather it has been

omitted from Table 1. Investigation is under way to establish from past *U. S. Weather Bureau Records* the coincident low temperature and wind conditions, including velocity and direction, which should be used in computing the design load for a heating system.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not be allowed to affect the size of the installation at all, although they may have a marked effect on the operation and control of the system. In general, it is safe to say that where audiences are involved, the heating installation must have sufficient capacity to bring the building up to the stipulated inside temperature before the audience arrives. In industrial plants, quite a different condition exists, and heat sources, if they are always available during the period of human occupancy, may be substituted for a portion of the heating installation. In no case should the actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

TABLE 4. HEAT EQUIVALENTS OF VARIOUS SOURCES

Source	Heat Equivalent Btu per Hr
Machinery (Motor in Room)	$\frac{\text{Motor Horsepower}}{\text{Efficiency}} \times 2544$
Machinery (Motor Outside Room)	Motor Horsepower $\times$ 2544
Electric Lights, per Kilowatt	3413
Producer Gas, per (Cubic Foot) (Hour)	150
Manufactured Gas, per (Cubic Foot) (Hour)	535
Natural Gas, per (Cubic Foot) (Hour)	1000

Electric Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat, which is retained in the room if the product being manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In some mills this is the chief source of heating and it is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round. Table 4 shows the heat output equivalent of various sources of heat in a factory. For information concerning the heat supplied by persons. refer to data given in Chapter 12. For other appliances see Table 16, Chapter 15.

INTERMITTENTLY HEATED BUILDINGS

In the case of intermittently heated buildings additional heat is required for raising the temperature of the air, the building materials and

the material contents of the building to the specified inside temperature. The rate at which this additional heat must be supplied depends upon the heat capacity of the structure and its material contents and upon the time in which these are to be heated[7].

This additional heat may be figured and allowed for as conditions require, but inasmuch as the heating system proportioned for taking care of the heat losses will usually have a capacity about 100 per cent greater than that required for average winter weather, and inasmuch as most buildings may either be continuously heated or have more time allowed for heating-up during the few minimum temperature days, no allowance is usually made except in the size of boilers or furnaces. For churches, auditoriums and other intermittently heated buildings, additional capacity should be provided.

RESIDENCE HEAT LOSS PROBLEMS

Example 5. Calculate the heat loss of residence shown in Fig. 2 located in the vicinity of Chicago. Assume inside and outside design temperatures to be 70 F and −10 F respectively. The attic is unheated. Assume ground temperature to be 50 F under basement and garage floors and 32 F adjoining basement walls. Estimate infiltration by crack method, assuming average wind velocity to be 12.5 mph during December, January and February. No wall, ceiling or roof insulation is to be figured in this problem, but all first and second floor windows are to have storm sash. The building is constructed as follows (transmission coefficients (*U*) in parentheses):

Walls: Brick veneer, building paper, wood sheathing, studding, metal lath and plaster (0.28). Walls of dormer over garage, same except wood siding in place of brick veneer (0.26).

Attic Walls: Brick veneer, building paper, wood sheathing on studding (0.42).

Basement Walls: 10 in. concrete (0.10).

Roof: Asphalt shingles on wood sheathing on rafters (0.53).

Ceiling (Second floor): Metal lath and plaster (0.69).

Windows: Double-hung wood windows with storm sash (0.45). Steel casement sash in basement (1.13).

Floor (Bedroom D): Maple finish flooring on yellow pine sub-flooring; metal lath and plaster ceiling below (0.25).

Floor (Basement and Garage): 4 in. stone concrete on 3 in. cinder concrete (0.10).

Solution: The calculations for this problem are given in Table 5, and a summary of the results in Table 6. The values in column F of Table 5 were obtained by multiplying together the figures in columns C, D, and E. The heat losses are calculated to the nearest 10 Btu. See reference notes for Table 5 for further explanation of data.

Attention is called to the summary of heat losses (Table 6) of the uninsulated residence (Fig. 2). As storm windows are used in this instance the *glass* and *door* transmission heat losses of 20.9 per cent are relatively small. The infiltration losses (12.8 per cent) are also comparatively small in this case because the storm windows serve substantially the same purpose as weatherstripping. In this problem, the wall, ceiling, and floor transmission losses comprise 66.3 per cent of the total.

Example 6. Calculate the heat loss of residence shown in Fig. 2 based on the same conditions as in *Example 5* but having construction improved or insulated to obtain coefficients as follows:

Walls, 0.13; *Walls of Dormer over Garage,* 0.12; *Attic Walls,* 0.28; *Walls Adjoining Garage,* 0.18; *Basement Walls (Recreation Room),* 0.10.

Roof, 0.53.

Ceiling (Second Floor), 0.15.

Windows (Same as in *Example 5*).

Floor (Bedroom D), 0.18.

Solution: The procedure for calculating the heat losses is similar to that for *Example 5.* A summary of the results is given in Table 7.

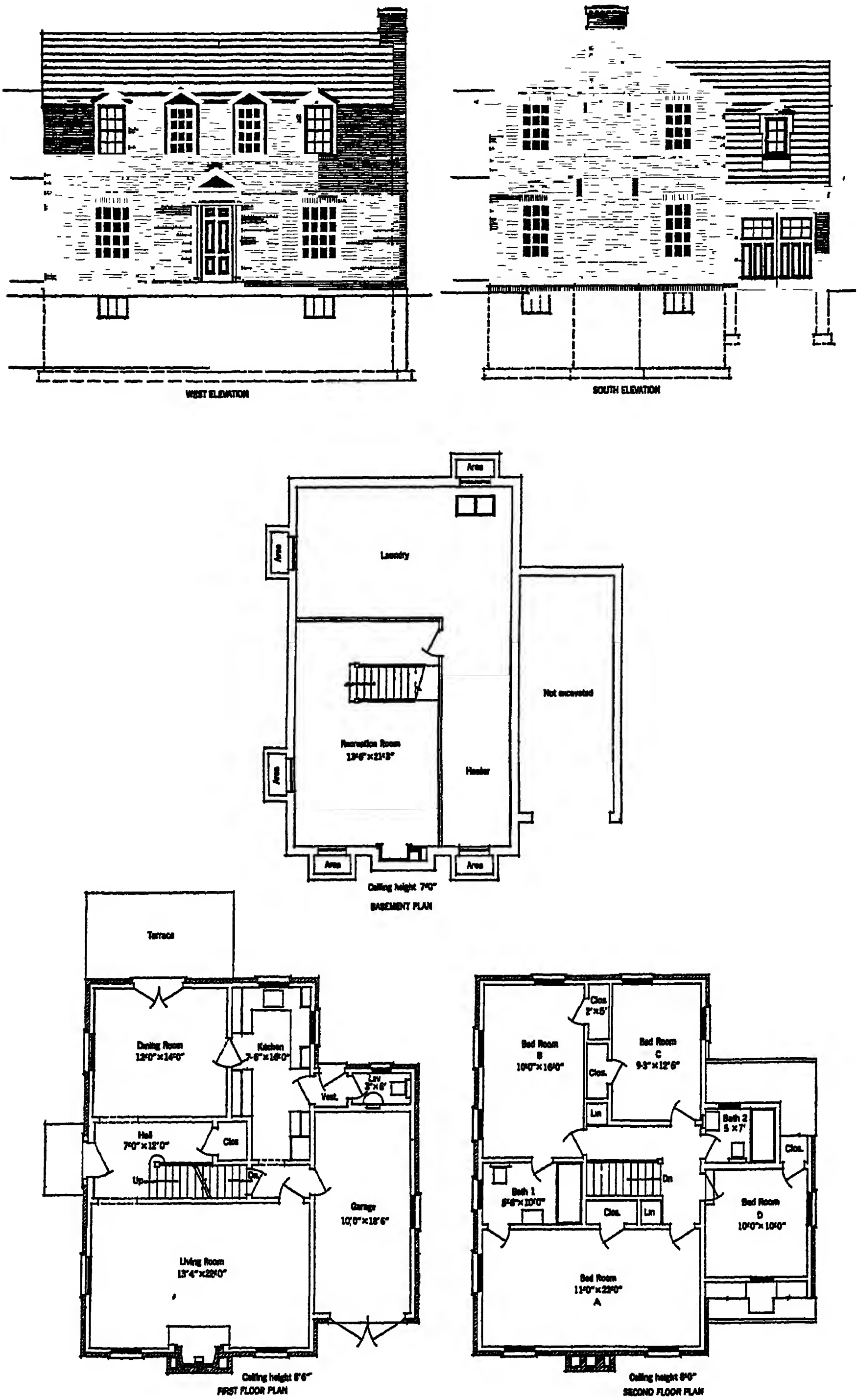

FIG. 2. ELEVATIONS AND FLOOR PLANS OF RESIDENCE

TABLE 5. HEAT LOSS CALCULATION SHEET FOR UNINSULATED RESIDENCE (FIG. 2)

A	B	C	D	E	F	G
Room or Space	Part of Structure	Net Area or Crack Length	Coefficient	Temp. Diff.[a]	Heat Loss (Btu per hour)	Totals (Btu per hour)
Bedroom A	Walls	238 sq ft	0.28	80	5330	
	Glass	40 sq ft	0.45	80	1440	
	Infiltration	36 lin ft[b]	0.35[c]	80	1010	
	Ceiling[d]	242 sq ft	0.69	39.8	6660	14,440
Bedroom B and Closet	Walls	156 sq ft	0.28	80	3490	
	Glass	40 sq ft	0.45	80	1440	
	Infiltration	36 lin ft[e]	0.35	80	1010	
	Ceiling[d]	160 sq ft	0.69	39.8	4400	10,340
Bedroom C	Walls	114 sq ft	0.28	80	2560	
	Glass	27 sq ft	0.45	80	970	
	Infiltration	18 lin ft[f]	0.35	80	500	
	Ceiling[d]	120 sq ft	0.69	39.8	3300	7,330
Bedroom D and Closet	Walls	118 sq ft	0.28	80	2650	
	Glass	20 sq ft	0.45	80	720	
	Infiltration	18 lin ft	0.35	80	500	
	Ceiling[d]	120 sq ft	0.69	39.8	3300	
	Floor over Garage	110 sq ft	0.25	35[g]	960[m]	8,130
Bathroom 1	Walls	30 sq ft	0.28	80	670	
	Glass	14 sq ft	0.45	80	500	
	Infiltration	18 lin ft	0.35	80	500	
	Ceiling[d]	55 sq ft	0.69	39.8	1510	3,180
Bathroom 2	Walls	79 sq ft	0.26	80	1770	
	Glass	9 sq ft	0.45	80	320	
	Infiltration	15 lin ft	0.35	80	420	
	Ceiling[d]	35 sq ft	0.69	39.8	960	
	Floor over Garage	35 sq ft	0.25	35	310[m]	3,780
Living Room	Walls	267 sq ft	0.28	80	5980	
	Walls (adjoining garage)	94 sq ft	0.39[h]	35	1280[m]	
	Glass	50 sq ft	0.45	80	1800	
	Infiltration	40 lin ft	0.35	80	1120	10,180
Dining Room	Walls	166 sq ft	0.28	80	3720	
	Glass (doors)	35 sq ft	1.13	80	3160	
	Glass (window)	20 sq ft	0.45	80	720	
	Infiltration[i]	31 lin ft	0.35	80	870	8,470
Kitchen and Entrance to Garage	Walls (outside)	96 sq ft	0.28	80	2150	
	Walls (adjoining garage)	51 sq ft	0.39[h]	35	700[m]	
	Infiltration	27 lin ft	0.35	80	760	
	Glass	18 sq ft	0.45	80	650	
	Door to garage	17 sq ft	0.51	35	300[m]	4,560
Lavette and Vestibule	Walls (outside)	82 sq ft	0.28	80	1840	
	Walls (adjoining garage)	85 sq ft	0.39[h]	35	1160[m]	
	Door	19 sq ft	0.51	80	780	
	Glass	9 sq ft	0.45	80	320	
	Infiltration	19 lin ft	0.35	80	530	4,630
Entrance Hall	Walls	39 sq ft	0.28	80	870	
	Door	21 sq ft	0.38	80	640	
	Infiltration	20 lin ft	0.35	80	560	
	Ceiling[d, p]	87 sq ft	0.69	39.8	2490	4,560
Garage	Walls	167 sq ft	0.28	45	2110	
	Glass	53 sq ft	1.13	45	2700	
	Doors	44 sq ft	0.51	45	1010	
	Infiltration	37 lin ft	1.62[j]	45	2700	
	Floor (heat gain)	185 sq ft	0.10[k]	−15[k]	−280	
	Heat gain				−4710[m]	3,530
Recreation Room[n]	Floor	287 sq ft	0.10	20	570	
	Walls	220 sq ft	0.10	38	840	
	Glass	8 sq ft	1.13	80	720	
	Infiltration	8 lin ft	0.76	80	490	2,620
					Total	85,750

Notes for Table 5.

[a]The inside-outside temperature difference is 70 — (−10) or 80 F, except where otherwise noted

[b]Only the south windows are used for arriving at the window crack for this room, on the assumption that whatever air enters through the south window cracks will leave through the west window cracks or elsewhere.

[c]Double-hung wood windows with storm sash are assumed to have the same leakage per foot of crack as weatherstripped windows. The air leakage per foot of crack is about 19.5 cu ft per foot of crack for a wind velocity of 12.5 mph. (See Table 2, Chapter 8.) The heat equivalent of the air leakage per (hour) (degree temperature difference per foot of crack) is obtained by multiplying this value by 0.018, or 19.5 × 0.018 = 0.35.

[d]In this problem the ceiling heat losses are calculated by estimating the attic temperature and then calculating the loss through the ceiling using the proper temperature difference. This unheated attic is not ventilated during the winter months. The attic temperature is estimated from Equation 4 to be 30.2 F when the outside temperature is −10 F and the room temperature is 70 F. The temperature difference is therefore 70 — 30.2 or 39.8 F.

[e]The window crack in the west wall having two windows is used.

[f]One-half the total crack is used in these rooms.

[g]Temperature in garage assumed to be 35 F.

[h]Coefficient for wall adjoining garage calculated on basis of metal lath and plaster on both sides of studs. (U = 0.39)

[i]The door crack is used for estimating the infiltration in this room and as the French doors are weatherstripped the infiltration coefficient is assumed to be the same as in Note b.

[j]The leakage for the garage doors is assumed to be twice that for poorly-fitted double-hung wood windows or about 90 cu ft per foot of crack for a wind velocity of 12 5 mph. The infiltration coefficient is therefore 0 018 × 90 or 1 62.

[k]The ground temperature is assumed to be 50 F and, as the garage temperature is 35 F, the heat transfer will be from the ground to the garage, and this heat gain should therefore be subtracted from the heat loss.

[m]The heat losses from various rooms into the garage are heat gains for the garage.

[n]Heat is to be provided for the recreation room and this space is therefore figured on the basis of a 70 F temperature. Heat loss into the basement from recreation room is neglected, the calculations being based only on losses through the outside walls, glass and floor.

[p]The upstairs hall ceiling is included with the downstairs entrance hall because these are connected by means of the stairway. The heat should be provided downstairs.

Table 6. Summary of Heat Losses of Uninsulated Residence

Heat losses given in Btu per hour

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	5330	6660	....	1440	1010	14,440
Bedroom B	3490	4400	....	1440	1010	10,340
Bedroom C	2560	3300	....	970	500	7,330
Bedroom D	2650	3300	960	720	500	8,130
Bathroom 1	670	1510	....	500	500	3,180
Bathroom 2	1770	960	310	320	420	3,780
Living Room	7260	....	....	1800	1120	10,180
Dining Room	3720	....	....	3880	870	8,470
Kitchen	2850	....	....	950	760	4,560
Lavette	3000	....	....	1100	530	4,630
Entrance Hall	870	2490	....	640	560	4,560
Garage	−1030[a]	....	−1550[b]	3410	2700	3,530
Recreation	840	....	570	720	490	2,620
Design Totals	33,980	22,620	290	17,890	10,970	85,750
Operating Totals[c]	33,980	22,620	290	17,890	5,485	80,265
Percentages[d]	42.3	28.1	0.4	22.3	6.8	100.0

[a]Wall heat loss of 2110 Btu minus wall heat gain of 3140 Btu.
[b]Heat gains: 960, 310 and 280 Btu.
[c]*Based on ½ computed infiltration.*
[d]Based on Operating Totals

TABLE 7. SUMMARY OF HEAT LOSSES OF INSULATED RESIDENCE

Heat losses given in Btu per hour

ROOM OR SPACE	WALLS	CEILING AND ROOF	FLOOR	GLASS AND DOOR	INFILTRATION	TOTALS
Bedroom A	2670	2370	....	1440	1010	7,490
Bedroom B	1750	1570	....	1440	1010	5,770
Bedroom C	1280	1170	....	970	500	3,920
Bedroom D	1320	1170	690	720	500	4,400
Bathroom 1	340	540	....	500	500	1,880
Bathroom 2	820	340	220	320	420	2,120
Living Room	3580	....	....	1800	1120	6,500
Dining Room	1860	....	....	3880	870	6,610
Kitchen	1400	....	....	950	760	3,110
Lavette	1460	....	....	1100	530	3,090
Entrance Hall	440	850	....	640	560	2,490
Garage	−400[a]	....	−1190[b]	3410	2700	4,520
Recreation	840	....	570	720	490	2,620
Design Totals	17,360	8,010	290	17,890	10,970	54,520
Operating Totals[c]	17,360	8,010	290	17,890	5,485	49,035
Percentages[d]	34.5	16.3	0.6	36.5	11 2	100.0

[a]Wall heat loss of 1050 Btu minus wall heat gains of 590, 320 and 540 Btu.
[b]Heat gains· 690, 220 and 280 Btu.
[c]Based on ½ computed infiltration.
[d]Based on Operating Totals.

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[6]—Measurements of Heat Losses from Slab Floor, by R. S. Dill, W. C. Robinson and H E. Robinson (*U. S Department of Commerce, National Bureau of Standards*, Building Materials and Structures Report BMS 103).

[7]—Heat Requirement Tables for Intermittently Heated Buildings, (*Engineering Experiment Station Bulletin*, No 60, *A. and M. College of Texas*, College Station, Texas), contains a set of tables applicable to either intermittent heating or cooling Further information may be found in a paper, A Method of Compiling Tables for Intermittent Heating, by Elmer G. Smith (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1942, p. 386).

CHAPTER 15

Cooling Load

Design Temperatures, Components of Heat and Water Vapor Gains, Solar Radiation, Instantaneous Heat Gain From Glass, Periodic Heat Flow Through Walls and Roofs, Heat Emission of Occupants, Heat Gain from Outdoor Air, Heat Emission of Appliances, Moisture Through Walls, Example of Cooling Load Estimate

LOAD calculations for summer air conditioning are more complicated than heating load calculations. Due to the variable nature of some of the contributing load components and the fact that they do not necessarily impose their maximum effect simultaneously, considerable care must be used in determining their phase relationship.

DESIGN TEMPERATURES

The conditions to be maintained in an enclosure depend upon several factors, especially the outside design conditions, duration of occupancy, and relationship between air motion, dry-bulb and wet-bulb temperatures. Information concerning the proper indoor effective temperature to be maintained is given in Chapter 12, for different geographical locations and for various age groups of individuals. Typical commercial design room conditions for the summer average peak load are shown in Table 1.

Summer dry-bulb and wet-bulb temperatures of various cities are given in Table 2. The temperatures are not the maximums but the design temperatures which are suggested for air conditioning calculations. The maximum outside wet-bulb temperatures as given in Weather Bureau reports usually occur only from 1 to 4 per cent of the time, and because they are of such short duration it is not practicable to design a cooling system for them. The temperatures shown in Table 2 are based on available design conditions known to be applied successfully.

COMPONENTS OF HEAT AND WATER VAPOR GAINS

The sources of the gain of heat within a building are:

1. Heat transferred through glass.

2. Heat transferred, usually as periodic heat flow, through the building materials that do not directly transmit incident solar radiation.

3. Heat emitted by occupants within the enclosure.

4. Heat added due to infiltration of warm outdoor air and escape of cooler room air.

5. Heat emitted by mechanical, chemical, gas, steam, hot water, and electrical appliances located within the enclosure.

The sources of gain of water vapor within a building are:

1. Water vapor emitted by occupants.

2. Water vapor added due to infiltration of outdoor air of higher humidity ratio and escape of room air of lower humidity ratio.

3. Water vapor emitted by vaporization processes within the enclosure.

When there is an addition of heat to the enclosure, as is the case, for example, with direct transfer of solar radiation, periodic heat flow, and electric lighting, the rate of such addition of heat in Btu per hour is often called a component of the *sensible heat gain*. When there is an addition

TABLE 1. TYPICAL COMMERCIAL DESIGN ROOM CONDITIONS FOR SUMMER AVERAGE PEAK LOAD[a]

TYPE OF INSTALLATION	DRY-BULB TEMP	WET-BULB TEMP	RELATIVE HUMIDITY PER CENT	GRAINS PER LB	EFFECTIVE TEMP
Deluxe Application	78	65	50	72.7	72.2
Normal Application	80	67	51	78.5	74.0
15 to 40 min Occupancy	82	68	49	80.0	75.3

[a]Values in Table 1 are for peak load conditions It is general practice to operate a system at approximately 76 F and 50 per cent relative humidity at other than peak load.

of water vapor to the air of the enclosure, the gain is often called a component of the *latent heat gain*. For example, when the humidity ratio of the air in a sealed enclosure is increased by water vapor emitted by human occupants, or by water vapor resulting from cooking processes, the heat required in both cases to vaporize the water does not come from the air. To maintain a constant humidity ratio in this sealed enclosure would require the condensation of water vapor in the cooling apparatus at a rate equal to the rate of its addition within the enclosure; the quantity of heat which would be removed in this process of condensation would be substantially equal to the product of the weight of water vapor emitted per hour and the latent heat of condensation of that vapor; this product, expressed in Btu per hour, would be called a latent heat gain.

Any simultaneous addition of heat and water vapor constitutes a source of both sensible heat gain and latent heat gain. For example, human occupants add both heat and water vapor, and the heat necessary to vaporize the water comes from the body rather than from the room air. Infiltration of outdoor air with a high dry-bulb temperature and a high humidity ratio and the accompanying escape of room air at a lower dry-bulb temperature and a lower humidity ratio constitute a source of both sensible heat gain and latent heat gain. Also, positive supply of outdoor air through the air conditioning apparatus and the exhaust of room air will, under the conditions just named, be a source of sensible heat gain and latent heat gain, although it may be advantageous in the analysis of the cooling load to separate gains of heat within the conditioned space from apparatus heat gains.

SOLAR RADIATION

Solar radiation is received on a plane perpendicular to the rays of the sun (normal incidence) outside the earth's atmosphere at the rate of about 420 Btu per (hour) (square foot). Before this radiation reaches a plane of normal incidence at the surface of the earth, part is scattered by air, water vapor, and dust. Such scattered radiation, or sky radiation, is not changed appreciably in wave length but will then be incident upon building surfaces which do not receive direct solar radiation at the same time, a familiar example being sky radiation incident upon north vertical glass in the middle of the day in north latitudes. Also some constituents of the earth's atmosphere, notably water vapor, carbon dioxide, and ozone absorb radiant energy. The total radiation, made up of direct and sky radiation, which reaches a plane perpendicular to the sun's rays at the earth's surface is less than 420 Btu per (hour) (square foot).

Standard values of the *direct* solar radiation incident upon a plane perpendicular to the sun's rays at the earth's surface suitable for many engineering calculations have been proposed by P. Moon [1]. These values

Table 2. Suggested Design Dry-Bulb, and Wet-Bulb Temperatures and Wind Velocities for Period June Through September[a]

Col. A State	Col. B City	Col. C Ground Elevation at Weather Station Ft	Col. D Design Dry-Bulb F	Col. E Design Wet-Bulb F	Col. F Avg Wind Velocity mph[b]
Ala.	Birmingham	694	95	78	5.2
	Mobile	10	92	79	8.6
	Montgomery	201	95	78	——
Ariz.	Flagstaff	6902	100	70	——
	Phoenix	1112	108	75	6.0
	Yuma	138	105	75	——
Ark.	Bentonville	1295	100	78	——
	Fort Smith	449	100	78	12.0
	Little Rock	257	98	79	6.0
Calif.	Fresno	277	104	70	7.9
	Los Angeles	312	90	70	6.0
	Needles	480	102	70	——
	Red Bluff	305	100	68	——
	Sacramento	25	105	74	5 0
	San Francisco	52	90	65	11.0
Colo.	Denver	5221	92	64	6 9
	Durango	6552	95	68	——
	Grand Junction	4587	96	65	6.1
	Pueblo	4799	95	70	——
Conn.	Hartford	15	93	76	——
	New Haven	13	95	75	7.3
Del.	Wilmington	70	95	78	9.7
D. C.	Washington	72	95	78	5.2
Fla.	Apalachicola	14	95	78	——
	Jacksonville	18	95	79	8.6
	Key West	6	95	81	——
	Miami	8	91	79	6.4
	Tampa	24	94	79	7.0
Ga.	Atlanta	987	95	77	8.5
	Augusta	134	95	78	——
	Macon	330	95	75	——
	Savannah	38	95	78	7.8
	Thomasville	276	95	78	——
Idaho	Boise	2842	97	65	5.9
	Lewiston	738	95	65	——
	Pocatello	4468	95	65	——
Ill.	Cairo	314	100	78	——
	Chicago	594	95	75	10.0
	Peoria	603	91	75	8.2
	Springfield	602	95	76	——
Ind.	Evansville	385	95	78	6.9
	Fort Wayne	777	95	75	——
	Indianapolis	718	95	76	9.0
Iowa	Des Moines	800	95	77	6.6
	Dubuque	641	95	75	——
	Keokuk	574	95	78	——
	Sioux City	1093	95	76	——
Kansas	Concordia	1375	95	78	——
	Dodge City	2522	95	78	——
	Wichita	1372	104	75	11.4
	Topeka	926	100	78	——
Ky.	Louisville	539	95	76	8.0
	Lexington	975	95	78	——
La.	New Orleans	3	92	80	6.2
	Shreveport	174	97	79	5.8
Maine	Eastport	33	90	73	——
	Greenville	1057	90	70	——
	Portland	61	90	73	7.3
Md.	Baltimore	14	95	78	6.9
Mass.	Boston	12	87	72	12.5
	Nantucket	43	90	75	——
Mich.	Alpena	587	93	73	——
	Detroit	619	93	73	10.2
	Houghton	695	93	73	——
	Ludington	690	95	75	——
	Marquette	674	93	73	——
	Sault Ste. Marie	607	93	73	——
Minn.	Duluth	1128	93	73	——
	Minneapolis	830	93	73	8.4
Miss.	Meridian	343	95	78	6.2
	Vicksburg	234	95	78	——
Mo.	Columbia	733	100	78	——
	Kansas City	741	102	76	9.5
	St. Louis	465	100	76	9.4
	Springfield	1265	93	75	8.7
Mont.	Billings	3568	95	70	——
	Havre	2488	95	68	——
	Helena	3893	89	62	8.0
	Kalispell	2956	95	68	——
	Miles City	2351	95	70	——
Nebr.	Lincoln	1184	95	75	9.3
	North Platte	2783	98	72	8.8
	Valentine	2581	95	75	——

TABLE 2. SUGGESTED DESIGN DRY-BULB AND WET-BULB TEMPERATURES AND WIND VELOCITIES FOR PERIOD JUNE THROUGH SEPTEMBER[a] (CONCLUDED)

COL. A STATE	COL. B CITY	COL. C GROUND ELEVATION AT WEATHER STATION FT	COL. D DESIGN DRY-BULB F	COL. E DESIGN WET-BULB F	COL. F AVG WIND VELOCITY MPH[b]
Nev.	Reno	4397	95	62	7.4
	Tonopah	6087	95	65	—
	Winnemucca	4288	95	65	—
N. H.	Concord	399	90	73	5 6
N. J.	Atlantic City	8	95	78	—
N. M.	Albuquerque	5314	94	66	7.6
	Roswell	3568	98	72	—
	Santa Fe	6312	90	65	6.5
N. Y.	Albany	119	89	72	7.1
	Buffalo	693	87	72	12.2
	Canton	406	95	75	12.9
	New York	10	91	75	12.9
	Oswego	292	93	73	—
	Syracuse	399	93	73	—
N. C	Asheville	2203	90	75	5.6
	Charlotte	753	95	76	—
	Manteo	12	95	78	—
	Raleigh	400	94	77	5.9
	Wilmington	6	90	78	6.9
N. Dak.	Bismarck	1650	95	73	8 8
	Devils Lake	1472	95	73	—
	Grand Forks	832	95	73	—
	Williston	1877	95	72	—
Ohio	Cincinnati	761	95	77	5 6
	Cleveland	787	93	73	11.2
	Columbus	724	95	77	—
Okla.	Broken Arrow	765	100	76	—
	Oklahoma City	1280	100	76	10 1
Ore.	Baker	3445	95	70	—
	Medford	1314	90	65	—
	Portland	30	89	66	6.6
Pa.	Erie	654	95	74	—
	Harrisburg	335	95	75	—
	Philadelphia	26	95	78	9.7
	Pittsburgh	1248	91	73	9.0
	Scranton	746	95	75	—
R. I.	Providence	12	93	75	10.0
S. C.	Charleston	9	92	79	9.9
	Columbia	332	95	78	—
S. Dak.	Huron	1282	99	72	10.0
	Pierre	1719	95	75	—
	Rapid City	3215	96	67	7.3
Tenn.	Chattanooga	689	95	77	6.5
	Knoxville	949	93	75	5.8
	Memphis	269	98	79	7.4
	Nashville	585	98	78	—
Texas	Abilene	1752	100	78	—
	Amarillo	3590	97	70	11.5
	Brownsville	16	92	79	7.0
	Corpus Christi	10	100	80	—
	Del Rio	957	100	75	—
	El Paso	3710	97	69	8.6
	Ft. Worth	688	100	78	9 4
	Galveston	7	95	80	9.7
	Palestine	492	100	78	—
	San Antonio	782	98	77	7.3
Utah	Modena	5460	92	63	—
	Salt Lake City	4222	97	63	8.2
Vt.	Burlington	398	90	73	8 9
Va.	Lynchburg	631	95	75	—
	Norfolk	11	95	78	10.9
	Richmond	162	95	78	6.2
	Wytheville	2299	95	76	—
Wash.	Seattle	14	80	65	8.1
	Spokane	1955	93	62	6 5
	Tatoosh	101	85	65	—
	Walla Walla	949	90	70	—
	Yakima	1068	90	68	—
W. Va.	Elkins	1969	95	75	—
	Parkersburg	615	95	75	5.3
Wis.	Green Bay	593	91	73	9.1
	La Crosse	664	93	76	6.4
	Milwaukee	674	95	75	10.4
	Wausau	1221	95	75	—
Wyo.	Cheyenne	6139	95	65	9.2
	Lander	5352	95	65	—
	Yellowstone Park	6239	95	65	—

[a]Compiled from U. S. Weather Bureau Records and various other sources.

[b]Blank spaces indicate data not available.

TABLE 3. PROPOSED STANDARD VALUES OF I_n, DIRECT SOLAR RADIATION RECEIVED AT NORMAL INCIDENCE, AT THE EARTH'S SURFACE

Solar Altitude β Deg	I_n Btu per (Hr) (Sq Ft)	Solar Altitude β Deg	I_n Btu per (Hr) (Sq Ft)	Solar Altitude β Deg	I_n Btu per (Hr) (Sq Ft)
5	65	30	234	60	276
10	122	35	245	70	283
15	165	40	253	80	289
20	196	45	260	90	294
25	219	50	266		

are calculated using the following assumptions: barometric pressure of 760 mm Hg (29.921 in.); depth of precipitable water of 20 mm (0.787 in.); dust particles, by counting, 300 per cc; partial pressure of the ozone layer in the atmosphere of 2.8 mm Hg (0.110 in.). The direct solar radiation received at normal incidence at the earth's surface depends, then, upon the altitude of the sun above a horizontal plane, and proposed standard values adapted from Moon's data are given in Table 3. Many of the data on solar radiation that follow are useful, not only in estimating cooling loads, but also in making calculations for solar heating.

The ratio of the direct solar radiation to the sky radiation received upon a *horizontal* plane at the surface of the earth depends upon solar altitude, time of year, cloudiness, and locality. Data are meager, but Table 4 gives approximate values of such ratios on clear days during the summer months in eastern states as a function of solar altitude alone.

INSTANTANEOUS HEAT GAIN FROM GLASS

When solar radiation, either direct or scattered, is incident upon glass, a fraction of the total (the transmissivity) is transmitted directly, a fraction (the absorptivity) is absorbed, and the remaining fraction (the reflectivity) is reflected. Under conditions of steady heat flow with negligible storage of heat in the glass, the energy absorbed by the glass must be transferred by convection and radiation at either or both of its surfaces.

If the temperature of the various surfaces of the enclosure and its furnishings be assumed equal to the temperature of the indoor air and if the thermal resistance of the glass be ignored, then heat is transferred by convection from the inside surface of glass to indoor air and by radiation to the surrounding solid surfaces exposed within the enclosure at the instantaneous rate of:

$$\left(\frac{q}{A}\right)_1 = U_g\,(t_a - t_i) + \frac{U_g}{f_o}\,(a_d\,I_d + a_s\,I_s) \qquad (1)$$

TABLE 4. APPROXIMATE RATIO OF DIRECT SOLAR RADIATION TO SKY RADIATION RECEIVED ON A HORIZONTAL SURFACE ON CLEAR DAYS IN EASTERN STATES

Solar Altitude β Deg	Ratio	Solar Altitude β Deg	Ratio	Solar Altitude β Deg	Ratio
0	0	40	3.84	70	5.63
10	1.40	50	4.55	80	5.90
20	2.30	60	5.20	90	6.10
30	3.10				

where

$\left(\frac{q}{A}\right)_1$ = Instantaneous rate of heat transfer from inside surface of glass of heat conducted through glass and radiation absorbed by glass, Btu per (hour) (square foot).

U_g = Over-all coefficient of heat transfer of glass, Btu per (hour) (square foot) (Fahrenheit degree).

t_a = Temperature of outdoor air, Fahrenheit degrees.

t_i = Temperature of indoor air, Fahrenheit degrees.

f_o = Film coefficient of heat transfer of outdoor air, Btu per (hour) (square foot) (Fahrenheit degree).

a_d = Absorptivity of glass for direct solar radiation, dimensionless.

a_s = Absorptivity of glass for sky radiation, dimensionless.

I_d = Direct solar radiation incident upon glass, Btu per (hour) (square foot).

I_s = Sky radiation incident upon glass, Btu per (hour) (square foot).

The instantaneous rate of direct transmission of direct solar and sky radiation to the solid surfaces within the room which receive this radiation is:

$$\left(\frac{q}{A}\right)_2 = \tau_d I_d + \tau_s I_s \tag{2}$$

where

τ_d = Transmissivity of glass for direct solar radiation, dimensionless.

τ_s = Transmissivity of glass for sky radiation, dimensionless.

$\left(\frac{q}{A}\right)_2$ = Instantaneous rate of direct transmission of solar and sky radiation through glass, Btu per (hour) (square foot).

$\frac{q}{A} = \left(\frac{q}{A}\right)_1 + \left(\frac{q}{A}\right)_2$ = Total instantaneous rate of heat gain from glass, Btu per (hour) (square foot).

Parmelee [2] has evaluated the effects of the controlling variables upon heat gain from glass. For a single sheet of common window glass with an index of refraction of 1.526 and a product of extinction coefficient and thickness of 0.054, Parmelee found $\tau_s = 0.778$ and $a_s = 0.06$; although the absorptivity for direct solar radiation varies with angle of incidence, for this glass a_d is substantially constant and equal to 0.08. The transmissivity for direct solar radiation of this glass, τ_d, depends upon angle of incidence with values interpolated from Parmelee's results as shown in Table 5.

For one thickness of common window glass, the instantaneous rate of heat transfer from the inside surface of the glass to the surroundings by convection and radiation becomes:

$$\left(\frac{q}{A}\right)_1 = 1.04\,(t_a - t_i) + 0.022\,I_d + 0.0165\,I_s \tag{1a}$$

For one thickness of common window glass, the instantaneous rate of

TABLE 5. TRANSMISSIVITY FOR DIRECT SOLAR RADIATION OF COMMON WINDOW GLASS

Angle of Incidence[a] Deg	Transmissivity τ_d	Angle of Incidence[a] Deg	Transmissivity τ_d	Angle of Incidence[a] Deg	Transmissivity τ_d
0	0.87	50	0.83	80	0.41
20	0.86	60	0.77	90	0
40	0.85	70	0.65		

[a]Of Direct Solar Radiation.

direct transmission of direct solar and sky radiation to the surfaces of the room on which such radiation is incident is:

$$\left(\frac{q}{A}\right)_2 = \tau_d\ I_d + 0.778\ I_s \qquad (2a)$$

Similar equations with other constants apply to glass of other qualities or in other arrangements, as heat-absorbing glass or glass in multiple layers.

The altitude of the sun and the angle of incidence of direct solar radiation on horizontal surfaces may be found when the latitude of the surface and the solar declination and hour angle are known; for vertical surfaces, the angle of solar incidence also depends, of course, upon the orientation of the surface [8]. The sky radiation incident upon a *vertical* surface may be assumed to be one-half of that incident upon a *horizontal* surface in the same latitude at the same time.

Example 1. Find the *instantaneous* rate of heat gain within an enclosure from a single thickness of horizontal, common, unshaded glass at a north latitude of 40 deg when the solar declination is 21.5 deg and the sun time is 3 p. m. The temperature of the outdoor air is 95 F and the temperature of the indoor air and room surfaces is 80 F. The solar altitude, β, may be found to be 47.7 deg, and the angle of incidence $(90 - \beta)$ of direct solar radiation upon the horizontal glass is 42.3 deg.

From Table 3, the direct solar radiation at normal incidence for β = 47.7 deg is I_n = 263 Btu per (hour) (square foot).

The direct solar radiation incident upon a horizontal surface is I_d = 263 cos (42.3 deg) = 194 Btu per (hour) (square foot).

The ratio of direct solar to sky radiation on a horizontal surface for β = 47.7 deg is found from Table 4 to be $\frac{I_d}{I_s} = 4.40$.

The sky radiation incident upon the horizontal surface is:

$$I_s = \frac{194}{4.4} = 44 \text{ Btu per (hour) (square foot).}$$

The instantaneous rate of heat gain by convection and radiation from the room surface of the glass is, from Equation 1a,

$$\left(\frac{q}{A}\right)_1 = 1.04\ (95 - 80) + 0.022\ (194) + 0.0165\ (44)$$
$$= 20.6 \text{ Btu per (hour) (square foot).}$$

The transmissivity of the glass for direct solar radiation for an angle of incidence of 42.3 deg is found from Table 5 as τ_d = 0.845.

The instantaneous rate of heat gain due to direct transmission of solar and sky radiation is found from Equation 2a as:

$$\left(\frac{q}{A}\right)_2 = 0.845\ (194) + 0.778\ (44) = 198.2 \text{ Btu per (hour) (square foot).}$$

The total instantaneous rate of heat gain from unshaded horizontal glass is:

$$\frac{q}{A} = \left(\frac{q}{A}\right)_1 + \left(\frac{q}{A}\right)_2 = 219 \text{ Btu per (hour) (square foot).}$$

Example 2. Same as *Example 1* but for west glass. The angle of incidence of the direct solar radiation upon west glass at the given time in the given locality may be found to be 48.95 deg. The direct solar radiation incident upon the west glass is:

$I_d = I_n$ cos (48.95 deg) = 263 (0.6567) = 173 Btu per (hour) (square foot).

The sky radiation incident upon the vertical glass may be taken as one-half of that incident upon horizontal glass at the same time, or

I_s = 0.5 (44) = 22 Btu per (hour) (square foot).

The instantaneous rate of heat gain by convection and radiation from the room surface of the glass is:

$$\left(\frac{q}{A}\right)_2 = 1.04\ (95 - 80) + 0.022\ (173) + 0.0165\ (22) = 19.8 \text{ Btu per (hour) (square foot).}$$

The transmissivity for direct solar radiation for an angle of incidence of 48.95 deg is

0.83 from Table 5, and the instantaneous rate of heat gain due to direct transmission of direct solar and sky radiation is:

$$\left(\frac{q}{A}\right)_2 = 0.83\ (173) + 0.778\ (22) = 160.7 \text{ Btu per (hour) (square foot).}$$

The total instantaneous rate of heat gain from the unshaded west glass is:

$$\left(\frac{q}{A}\right) = \left(\frac{q}{A}\right)_1 + \left(\frac{q}{A}\right)_2 = 181 \text{ Btu per (hour) (square foot).}$$

Heat Gain and Cooling Load

Some instantaneous rates of heat gain within an enclosure are not transmitted undiminished and without time lag into cooling loads on the air conditioning apparatus [4, 5]. By way of illustration, consider the instantaneous rate of heat gain from west glass in *Example 2*. The instantaneous rate of heat gain by convection and radiation from the room surface of the glass is substantially 20 Btu per (hour) (square foot); of this, about 7 Btu per (hour) (square foot) are being transferred by convection to the air of the enclosure and constitute an instantaneous cooling load. If the small amount of radiant energy which is absorbed by the water vapor and carbon dioxide in the air of the enclosure be ignored, the remainder of the total gain of heat from the glass (consisting of the 13 Btu per (hour) (square foot) transferred by radiation from the inside surface of the glass to ceiling, walls, floor and surfaces of furnishings in the room and the 161 Btu per (hour) (square foot) transmitted directly, principally to the floor or floor covering, does *not* constitute an immediate and undiminished load on the cooling apparatus.

The solid surfaces of the room which receive this radiant energy must absorb it and then transfer heat by convection to the room air before the load is felt by the air conditioning equipment. To make a simplifying assumption in further explanation of this process, it is assumed that all surfaces receiving this radiant energy within the room are warmed at the same rate; and it is assumed also that there are 20 sq ft of non-glass surface in the room per square foot of glass surface. To transfer the radiant energy that is being received by these surfaces by convection at the same rate to the room air would require that these surfaces be warmed to a temperature at least 15 deg higher than that of the air. Such warming requires time due to the heat capacity of the solid material involved, and while this material is being warmed, the instantaneous rate of heat gain within the enclosure is constantly changing. The process is one of unsteady heat flow with storage of heat, and the net effect is that the *cooling load* due to west glass at 3 p. m. will undoubtedly be some *fraction of the instantaneous rate of heat gain from west glass at an earlier time of day*.

There are other heat gains which have considerable components in the form of radiated energy. These include lighting, people, and apparatus. If the radiant component of the heat gain is steady over a long period of time, the difference between instantaneous heat gain and instantaneous cooling load on air conditioning apparatus is not significant. More study would be required to evaluate the time lag and the ratio of instantaneous cooling load to instantaneous rate of heat gain due to radiant heat emission from sources with fluctuating rates of such emission. At present, however, it seems reasonable to assume that this ratio is about 0.8 or 0.9.

The instantaneous total rate of heat gain from sunlit glass is given in Table 6 as calculated by the application of the principles previously explained. As the *values given are for a relatively clear atmosphere*, it may

Table 6. Total Instantaneous Rates of Heat Gain Through Single, Unshaded, Common Glass, for Variously Oriented Vertical and for Horizontal Positions[a,b]

Data for Solar Declination of 17.5 Deg—August 1

Sun Time	Solar Altitude β Deg	Total Instantaneous Rate of Heat Gain, Btu per Hour for Each Square Foot of Unshaded Glass								
		N	NE	E	SE	S	SW	W	NW	Horizontal
					25 Deg North Latitude					
6 a.m.	7.5	19	77	84	40	3	3	3	3	12
7	20.5	26	146	173	98	10	10	10	10	68
8	34.0	18	148	193	121	13	13	13	13	143
9	47.5	15	118	172	120	16	15	15	15	205
10	61.5	16	67	124	95	22	16	16	16	252
11	74.5	16	25	57	57	25	16	16	16	282
12	83 0	16	16	16	25	28	25	16	16	293
1 p.m.	74.5	16	16	16	16	25	57	57	25	282
2	61.5	16	16	16	16	22	95	124	67	252
3	47.5	15	15	15	15	16	120	172	118	205
4	34.0	18	13	13	13	13	121	193	148	143
5	20.5	26	10	10	10	10	98	173	146	68
6	7.5	19	3	3	3	3	40	84	77	12
					30 Deg North Latitude					
6 a.m.	9.0	22	88	97	47	4	4	4	4	15
7	21.5	23	146	176	105	11	11	11	11	74
8	34.5	16	140	194	130	14	14	14	14	144
9	47.5	15	104	171	133	20	15	15	15	205
10	60.0	16	53	126	112	33	16	16	16	248
11	72.0	16	19	56	74	42	16	16	16	277
12	78 0	16	16	16	34	45	34	16	16	288
1 p m.	72.0	16	16	16	16	42	74	56	19	277
2	60.0	16	16	16	16	33	112	126	53	248
3	47.5	15	15	15	15	20	133	171	104	205
4	34.5	16	14	14	14	14	130	194	140	144
5	21.5	23	11	11	11	11	105	176	146	74
6	9.0	22	4	4	4	4	47	97	88	15
					35 Deg North Latitude					
6 a.m.	10.0	21	97	109	53	4	4	4	4	19
7	22.5	19	143	179	110	11	11	11	11	79
8	34.5	14	133	194	140	15	14	14	14	144
9	46.5	15	90	170	144	28	15	15	15	200
10	58.5	16	38	126	128	47	16	16	16	243
11	68 5	16	16	56	91	62	17	16	16	269
12	73 0	16	16	16	45	68	45	16	16	279
1 p.m.	68.5	16	16	16	17	62	91	56	16	269
2	58.5	16	16	16	16	47	128	126	38	243
3	46.5	15	15	15	15	28	144	170	90	200
4	34 5	14	14	14	14	15	140	194	133	144
5	22 5	19	11	11	11	11	110	179	143	79
6	10.0	21	4	4	4	4	53	109	97	19
					40 Deg North Latitude					
5 a m	1 5	7	18	17	6	1	1	1	1	2
6	11 5	23	106	120	62	5	5	5	5	24
7	23.0	15	141	181	118	11	11	11	11	82
8	34.5	14	122	194	147	19	14	14	14	145
9	45 5	15	76	172	156	42	15	15	15	196
10	56.0	16	30	125	144	66	16	16	16	235
11	64.5	16	16	53	110	85	22	16	16	261
12	68 0	16	16	16	62	94	62	16	16	269
1 p.m.	64.5	16	16	16	22	85	110	52	16	261
2	56 0	16	16	16	16	66	144	125	30	235
3	45.5	15	15	15	15	42	156	172	76	196
4	34.5	14	14	14	14	19	147	194	122	145
5	23.0	15	11	11	11	11	118	181	141	82
6	11.5	23	5	5	5	5	62	120	106	24
7	1.5	7	1	1	1	1	6	17	18	2

TABLE 6. TOTAL INSTANTANEOUS RATES OF HEAT GAIN THROUGH SINGLE, UNSHADED, COMMON GLASS, FOR VARIOUSLY ORIENTED VERTICAL AND FOR HORIZONTAL POSITIONS[a,b]—(CONCLUDED)

Sun Time	Solar Altitude β Deg	Total Instantaneous Rate of Heat Gain, Btu per Hour for Each Square Foot of Unshaded Glass								
		N	NE	E	SE	S	SW	W	NW	Horizontal
					45 Deg North Latitude					
5 a.m.	2.0	9	23	23	8	1	1	1	1	2
6	12.5	22	111	129	68	6	6	6	6	28
7	23.0	13	135	182	121	11	11	11	11	82
8	33.5	14	116	192	153	24	14	14	14	141
9	44.0	15	63	168	166	55	15	15	15	189
10	53.0	16	22	123	154	88	16	16	16	225
11	60.0	16	16	56	127	113	30	16	16	249
12	63.0	16	16	16	76	119	76	16	16	256
1 p.m	60.0	16	16	16	30	113	127	56	16	249
2	53.0	16	16	16	16	88	154	123	22	225
3	44.0	15	15	15	15	55	166	168	63	189
4	33.5	14	14	14	14	24	153	192	116	141
5	23.0	13	11	11	11	11	121	182	135	82
6	12.5	22	6	6	6	6	68	129	111	28
7	2.0	9	1	1	1	1	8	23	23	2
					50 Deg North Latitude					
5 a.m.	4.5	18	48	48	17	2	2	2	2	5
6	13.5	22	119	139	75	6	6	6	6	32
7	23.5	11	131	183	127	11	11	11	11	84
8	33.0	13	103	190	161	30	13	13	13	138
9	42.0	14	51	165	173	69	14	14	14	179
10	50.0	16	19	122	166	109	16	16	16	214
11	56.0	16	16	55	138	133	41	16	16	235
12	58.0	16	16	16	92	140	92	16	16	242
1 p.m.	56.0	16	16	16	41	133	138	55	16	235
2	50.0	16	16	16	16	109	166	122	19	214
3	42.0	14	14	14	14	69	173	165	51	179
4	33.0	13	13	13	13	30	161	190	103	138
5	23.5	11	11	11	11	11	127	183	131	84
6	13.5	22	6	6	6	6	75	139	119	32
7	4.5	18	2	2	2	2	17	48	48	5

[a]Table compiled from solar radiation transmission data developed by A.S.H.V.E. Research Laboratory[2] and direct intensity values developed by Moon[1], recently enlarged and revised by the A.S.H.V.E. Research Laboratory.

[b]For relatively clear atmosphere at sea level. For hazy atmosphere values may be reduced 10 per cent. Above sea level add one per cent per 1000 ft elevation

be necessary to reduce the values by as much as 10 per cent where the atmosphere is hazy from industrial smoke or other causes. Also, *the values are for sea level* and approximate correction for altitude may be made by increasing the values one per cent for each thousand feet of altitude. In preparing these tables, the declination of the sun was taken as 17.5 deg (August 1); if the instantaneous rate of heat gain is desired for some declination of the sun other than 17.5 deg, use the latitude that differs from that of the locality by the difference of the declination from 17.5 deg. For example, assume that the instantaneous rate of heat gain from glass is required for Philadelphia in the middle of June; the solar declination in mid-June is about 23.5 deg, and the latitude of Philadelphia is 40 deg; the table for 45 deg north latitude will give approximately the correct results.

Heat Absorbent Glass

Values shown in Table 6 give the total instantaneous rates of heat gain through a single sheet of sunlit common glass. The extinction (or absorption) coefficients of so-called heat absorbing glasses are several

TABLE 7. RATIO OF TOTAL INSTANTANEOUS RATE OF HEAT GAIN THROUGH SINGLE THICKNESS OF SUNLIT GLASS TO HEAT GAIN FOR COMMON GLASS AS GIVEN IN TABLE 6

TRANSMISSIVITY[a]	APPROXIMATE RATIO	TRANSMISSIVITY[a]	APPROXIMATE RATIO	TRANSMISSIVITY[a]	APPROXIMATE RATIO
0.87	1.0	0.60	0.76	0.30	0.52
0.80	0.93	0.50	0.68	0.20	0 45
0.70	0.85	0.40	0.60		

[a]For direct solar radiation at normal incidence.

times as great as for common glass. As a consequence, the absorptivity of a heat-absorbing glass is greater and the transmissivity less than the values for common glass in the same thickness. In the preparation of Table 6, the transmissivity of common glass for direct solar radiation at normal incidence was taken as 0.87; for glasses with other transmissivities, the total instantaneous rate of heat gain may be multiplied by approximate factors which are given in Table 7.

Multiple Glass Layers

By using two or more layers of glass separated by air spaces, the absorptivity is increased and the transmissivity decreased. Values for some combinations appear in the work of Parmelee [2]. A rough approximation which holds fairly well until the angle of incident radiation becomes large is that the transmissivity of the combination is the product of the transmissivities of the component layers, while the absorptivity of the combination is the absorptivity of the outer glass plus the product of the absorptivity of the inner glass and the transmissivity of the outer.

Shading of Glass

The rate of gain of heat through sunlit glass may be considerably reduced by proper shading, and the possibility of such shading should be investigated whenever heat gain from glass constitutes a large portion of the cooling load.

Vertical glass which is not mounted in the plane of the building surface is partially shaded by the *setback*. If a vertical window of height l and

TABLE 8. VALUES OF ANGLE γ FOR AUGUST 1
(*For Use in Equation 8*)

NORTH LATITUDE DEG	SUN TIME	ANGLE γ—DEGREES		
		East Glass	South Glass	West Glass
30	9 a.m.	5	85	....
	12 noon	90	0	90
	3 p.m.	....	85	5
	6 p.m.	....	90	16
40	9 a.m.	16	74	....
	12 noon	90	0	90
	3 p.m.	....	74	16
	6 p.m.	....	90	14
50	9 a.m.	25	65	....
	12 noon	90	0	90
	3 p.m.	....	65	25
	6 p.m.	....	90	12

TABLE 9. EFFECT OF SHADING UPON TOTAL RATES OF INSTANTANEOUS HEAT GAIN THROUGH GLASS

TYPE OF SHADING	FINISH	FRACTION OF GAIN THROUGH UNSHADED WINDOW
Outside Shading Screen: metal slats 0.050 in. wide, spaced 0.063 in. apart and set at 17 deg angle with horizontal.	Dark	0.20-0 35
Canvas Awning	Dark	0.25-0.35
Outside Venetian Blind: slats at 45 deg, extended as an awning without sides to cover approximately two-thirds of window.	Light	0.35-0.50
Inside Roller Shade: fully drawn.	Aluminum	Approx. 0.45
Outside Venetian Blind: slats at 45 deg, fully covering window.	Aluminum	Approx. 0 6
Inside Venetian Blind: slats at 45 deg, fully covering window.	Aluminum	0.65-0.80
Inside Roller Shade: half drawn.	Buff	Approx. 0.7
Inside Roller Shade: half drawn.	Dark	0 90-0.95

width w be set back from the plane of the building a distance s, the fraction of the total area of the window which receives direct solar radiation is:

$$G_f = 1 - r_1 \tan \beta - r_2 \tan \gamma + r_1 r_2 \tan \beta \tan \gamma \tag{3}$$

where

$r_1 = s/l$, $r_2 = s/w$, β = solar altitude, and γ is the angle between the traces on a horizontal plane of the edge of the wall or frame providing shading and of the direct rays of the sun.

Solar altitudes are given in Table 6 for August 1 and various latitudes. Values of the angle γ for August 1 and three north latitudes are given in Table 8 for east, south, and west glass. Similar tables may be prepared for other dates of interest in solar heating studies.

Example 3. Estimate the instantaneous rate of heat gain from a west window 3 ft wide by 5 ft high with a setback of 6 in. for August 1 and a north latitude of 40 deg at 3 p. m. (sun time).

From Table 6, the instantaneous rate of heat gain per square foot of sunlit glass is 172 Btu per hour. From the same table, the solar altitude is 45.5 deg. From Table 8, the angle γ is 16 deg, from Equation 3, the fraction of the total window area that is receiving direct solar radiation is:

$$\begin{aligned} G_f &= 1 - 0.1 \tan 45.5 - 0.167 \tan 16 + 0.0167 \tan 45.5 \tan 16. \\ &= 1 - 0.102 - 0.048 + 0.005 = 0.855 \end{aligned}$$

Although the heat conducted through the glass and the sky radiation transmitted are not reduced in proportion to the shading from direct solar radiation, it is approximately correct to estimate the instantaneous rate of heat gain from this window as:

$q = 3 \times 5 \times 0.855(172) = 2206$ Btu per hour.

The desirability of providing additional shading for this window is obvious from the high rate of heat gain. Shading devices include awnings, shades, and screens. In Table 9 are given the ratios of the instantaneous rates of heat gain from windows shaded by different indoor and outdoor shading fixtures to the rate of heat gain from unshaded glass according to tests [6] at the A.S.H.V.E. Research Laboratory. There are a number of variables affecting these ratios such as color, fit, solar altitude, and angle of incidence of the solar radiation. These values, therefore, must be considered as approximate, only, and will have to be used with considerable judgment. An inside shade is effective to the extent of its

Table 10. Instantaneous Total Rates of Heat Gain Through Glass Blocks on August 1

Sun Time	Total Instantaneous Rate of Heat Gain, Btu per Hour for Each Square Foot of Sunlit Glass					
	Vertical Surface Facing					
	East	West	South			
North Latitude, Deg	30 to 45	30 to 45	30	35	40	45
7 a.m.	61	—	−4.5	−2.0	−0.5	1.0
8	78	—	0.0	2.0	4.0	5.0
9	74	5.0	5.0	7.0	10	12
10	58	6.5	11	15	18	21
11	45	7 5	17	22	26	32
12 noon	37	11	22	28	34	41
1 p.m.	30	22	25	32	39	46
2	24	35	26	32	39	47
3	20	55	24	30	37	45
4	16	77	20	26	32	41
5	13	86	15	20	25	34
6	11	55	9.5	14	18	26
7	8	19	3 5	7.0	11	18

reflectivity, for the portion of the solar radiation directly transmitted by the glass that is absorbed by the shade is transferred by convection to the room air and by radiation to the solid surfaces of the room.

The instantaneous total rates of heat gain through glass blocks are given in Table 10 for various times of day for south, east, and west exposures for different latitudes on August 1. The table is based upon a temperature of the indoor air of 78 F and for a design day having a maximum temperature of 95 F. The values are from data obtained by the A.S.H.V.E. Research Laboratory[7] and are averages for four typical glass block designs, two with smooth exterior faces and the other two with ribbed exterior faces.

PERIODIC HEAT FLOW THROUGH WALLS AND ROOFS

The flow of heat through building walls or roofs that do not transmit solar radiation directly is a complex process under the usual summer conditions where the temperature at a given point in the material does not remain constant with respect to time. Although the temperature of the indoor air may be held constant, the solar and sky radiation incident upon the weather side of the structure, as well as the temperature of the outdoor air, vary considerably with time, with the result that the instantaneous maximum rate of gain of heat within the enclosure lags behind the instantaneous maximum rate of entry of heat into the weather side of the material and may be considerably less than this rate of entry. In effect, the instantaneous rate of heat gain within the enclosure is some fraction of the instantaneous rate of heat entry on the weather side at an earlier time; the greater the heat capacity of the material, the greater is the time lag and the smaller is the ratio of rate of heat gain to rate of heat entry. Some results are available from A.S.H.V.E.—Cornell University cooperative research and other studies of the subject [8].

Sol-Air Temperature

Outdoor design conditions must be established before the rate of heat gain may be estimated. A convenient way of combining the effects of solar and sky radiation, solar absorptivity, temperature, and air move-

ment is to use the concept of sol-air temperature. For either steady or unsteady flow of heat, the instantaneous rate of heat entry into the weather side of a sunlit building material that does not transmit, directly, solar or sky radiation is:

$$\left(\frac{q}{A}\right)_L = bI + f_o\,(t_a - t_L) \text{ Btu per (hour) (square foot).}$$

where

b = Absorptivity of weather side of material for incident solar and sky radiation.
I = Rate of incidence of solar and sky radiation, Btu per (hour) (square foot).
t_L = Temperature of weather surface of the material, Fahrenheit degrees.

Let the sol-air temperature be defined as:

$$t_e = t_a + \frac{bI}{f_o} \qquad (4)$$

Then, the instantaneous rate of heat entry into the weather side of the structure is:

$$\left(\frac{q}{A}\right)_L = f_o\,(t_e - t_L)$$

The sol-air temperature, t_e, is the temperature of outdoor air which, in contact with the weather side of a material that was receiving no solar or sky radiation, would give the same rate of heat entry into that surface as exists with the actual combination of incident solar and sky radiation and temperature of the outdoor air. For example, if t_a = 90 F, b = 0.7, I = 200 Btu per (hour) (square foot), and f_o = 4, the sol-air temperature is:

$$t_e = 90 + \frac{0.7\,(200)}{4} = 125 \text{ F.}$$

TABLE 11. SUMMER DESIGN SOL-AIR TEMPERATURES FOR NEW YORK, N. Y. (NORTH LATITUDE 40°46′; ELEVATION 180 FT)

SUN TIME	SOL-AIR TEMPERATURE, t_e FAHRENHEIT DEGREES					
	Any Surface	Horizontal	North	East	South	West
Ratio $\frac{b}{f_o}$	0	0.25	0.25	0.25	0.25	0.25
12 midnight	79	79	79	79	79	79
1 a.m.	78	78	78	78	78	78
2	77	77	77	77	77	77
3	77	77	77	77	77	77
4	76	76	76	76	76	76
5	76	76	76	76	76	76
6	76	81	80	89	77	77
7	80	96	85	106	82	82
8	82	110	85	114	86	85
9	86	127	90	120	97	90
10	88	137	92	114	104	92
11	90	148	94	106	111	94
12 noon	92	155	97	97	115	98
1 p.m.	93	154	98	98	117	108
2	94	152	99	99	112	126
3	94	144	99	99	102	136
4	94	136	99	99	99	145
5	93	121	103	97	97	140
6	90	106	106	93	93	134
7	88	93	102	90	90	115
8	85	85	85	85	85	85
9	83	83	83	83	83	83
10	82	82	82	82	82	82
11	81	81	81	81	81	81
24-hr avg, t_m	84.8	106.4	88.5	92.3	90.7	96.5

In other words, the instantaneous rate of entry of heat into the weather side of this material is precisely the same as if the air temperature were 125 F with no solar and sky radiation incident upon the surface.

Data of the U. S. Weather Bureau for the 10-year period from 1932 through 1941 have been studied for New York, N. Y., [8], and Lincoln, Nebr.[9]. Only simultaneous values of the air temperature and solar and sky radiation have been combined in determining design values of the sol-air temperature for various surfaces at different times of day in these localities. Since the 24-hour average of the sol-air temperature is greater in July than for any other month at both stations, the sol-air temperature at each hour in July which was equalled or exceeded at that hour only 16 times in 310 observations was chosen as the *design* sol-air temperature.

Summer design sol-air temperatures are given in Table 11 for New York, N. Y.; Table 12 gives similar data for Lincoln, Nebr.

The solar absorptivity, b, of a surface depends principally upon color with the following approximate values suggested:

b = 0.4 for white and other light colors.

b = 0.7 for red and other medium colors.

b = 1.0 for black and very dark colors.

An example of the use of the tables in determining sol-air temperature is given.

Example 4. Find the summer design sol-air temperature for a west wall in New York, N. Y., which has a solar absorptivity of 0.7, at a sun time of 3 p. m.

$$\text{Use } f_o = 4\text{; then } \frac{b}{f_o} \text{ for this wall is } \frac{0.7}{4.0} \text{ or } 0.175.$$

TABLE 12. SUMMER DESIGN SOL-AIR TEMPERATURES FOR LINCOLN, NEBR. (NORTH LATITUDE 40°50′; ELEVATION 1225 FT)

SUN TIME	SOL-AIR TEMPERATURE, t_e, FAHRENHEIT DEGREES					
	Any Surface	Horizontal	North	East	South	West
Ratio: $\frac{b}{f_o}$	0	0.25	0 25	0.25	0.25	0.25
12 midnight	89	89	89	89	89	89
1 a.m.	88	88	88	88	88	88
2	86	86	86	86	86	86
3	84	84	84	84	84	84
4	84	84	84	84	84	84
5	82	82	82	82	82	82
6	81	87	88	100	82	82
7	82	103	93	125	85	85
8	88	124	94	137	92	92
9	93	143	98	142	104	98
10	96	160	102	138	115	102
11	100	172	106	129	125	106
12 noon	102	178	108	115	130	108
1 p.m.	104	180	110	110	132	119
2	106	178	112	112	131	137
3	107	170	113	113	126	150
4	107	158	112	112	117	158
5	106	142	113	110	110	157
6	105	126	117	108	108	149
7	102	109	113	104	104	128
8	98	99	98	98	98	98
9	94	94	94	94	94	94
10	92	92	92	92	92	92
11	90	90	90	90	90	90
24-hr avg. t_m	94.4	121 6	98.6	105.9	102.1	106.6

From Table 11: for $\frac{b}{f_o} = 0$, $t_e = 94$ F; for $\frac{b}{f_o} = 0.25$, $t_e = 136$ F; and by linear interpolation for this wall:

$$t_e = 94 + \frac{0.175}{0.25}(136 - 94) = 94 + 29 = 123 \text{ F.}$$

Homogeneous Walls or Roofs

For walls or roofs of a single, homogeneous material, the instantaneous rate of heat gain within the enclosure due to periodic heat flow when the temperature of the indoor air is held constant is, approximately,

$$\frac{q}{A} = U(t_m - t_i) + \lambda U(t'_e - t_m) \text{ Btu per (hour) (square foot)} \qquad (5)$$

where

t_m = 24-hr average sol-air temperature for the particular value of $\frac{b}{f_o}$, Fahrenheit degrees.

λ = A variable that depends upon the thickness, material, and orientation of the wall or roof; see Table 13 for values.

t'_e = Sol-air temperature at a time earlier than the time for which the heat gain is being found by an amount that is equal to the time lag of the wall or roof, Fahrenheit degrees; see Table 13 for values of time lag.

U = Over-all coefficient of heat transfer of the wall or roof, Btu per (hour) (square foot) (Fahrenheit degree).

$$\frac{1}{\frac{1}{f_i} + \frac{1}{f_o} + \frac{L}{k}} = \frac{1}{\frac{1}{1.65} + \frac{1}{4} + \frac{L}{k}} = \frac{1}{0.856 + \frac{L}{k}}$$

L = Thickness of building material, feet.

TABLE 13 PERIODIC HEAT FLOW DATA FOR HOMOGENEOUS WALLS OR ROOFS

Material	Thickness, In.	Over-all Coefficient, Btu per (hr) (sq ft) (°F) U[a]	Thermal Resistance of Solid Material, (hr) (sq ft) (°F)/Btu $\frac{L}{k}$	Time Lag, Hr	Factor, λ, in Equation 5			
					Horizontal and North	East	South	West
Stone	8	0.67	0.64	5.5	0.51	0 36	0.48	0.42
	12	0.55	0.96	8.0	0.28	0.19	0.26	0.22
	16	0.47	1.28	10.5	0.17	0.10	0.15	0.13
	24	0.36	1.92	15.5	0.06	0 03	0.05	0.04
Solid Concrete	2	0.98	0.17	1.1	0.93	0.87	0.92	0.89
	4	0.84	0.33	2 5	0.79	0.68	0.76	0.72
	6	0.74	0 50	3.8	0.61	0 46	0.58	0.51
	8	0.66	0 67	5 1	0.49	0.33	0.46	0.39
	12	0.54	1.00	7.8	0.29	0.17	0.26	0.22
	16	0.46	1.33	10.2	0.17	0.09	0.15	0.12
Common Brick	4	0.60	0.80	2 3	0.83	0 75	0.81	0.78
	8	0.41	1.60	5.5	0.51	0.39	0.49	0.44
	12	0.31	2.40	8 5	0.26	0.17	0.25	0.21
	16	0.25	3.20	12.0	0.13	0.08	0.12	0.10
Face Brick	4	0.77	0.44	2.4	0.81	0.70	0.78	0.74
Wood	½	0.68	0.62	0.17	1.0	1.0	1.0	1.0
	1	0.48	1 25	0.45	1.0	0.99	0.99	0.99
	2	0.30	2.50	1.3	0.98	0.91	0.96	0.94
Insulating Board	½	0.42	1.51	0.08	1.0	1.0	1.0	1 0
	1	0.26	3.03	0.23	1.0	1.0	1.0	1.0
	2	0.14	6.05	0.77	1.0	1 0	1.0	1 0
	4	0.08	12.1	2.7	0.83	0.74	0.81	0.76
	6	0.05	18.2	5.0	0.64	0.49	0.61	0.55

[a]Based upon an outdoor combined film coefficient of 4.0 and an indoor combined film coefficient of heat transfer of 1.65 Btu per (hour) (square foot) (Fahrenheit degree).

k = Thermal conductivity of building material, Btu per (hour) (square foot) (Fahrenheit degree per foot).

f_i = Film coefficient of heat transfer of indoor air Btu per (hour) (square foot) (Fahrenheit degree).

Examples in the use of Tables 11, 12, and 13 are given.

Example 5. Find the instantaneous design rate of heat gain through an 8-in. west wall of common brick ($b = 0.7, f_o = 4$) located in New York, N. Y., at 9:30 p. m. sun time, when the temperature of the indoor air is constant at 80 F.

From Table 13, $U = 0.41$, the time lag is 5.5 hr, and $\lambda = 0.44$.

By linear interpolation on the basis of the value of b/f_o in Table 11,

$$t_m = 84.8 + \frac{0.175}{0.25}(96.5 - 84.8) = 93 \text{ F.}$$

The design sol-air temperature at a time earlier than 9:30 p. m. by the time lag (at 4:00 p. m.) for a west wall in New York, N. Y., is, from Table 11,

$$t'_e = 94 + \frac{0.175}{0.25}(145 - 94) = 129.7 \text{ F.}$$

From Equation 5, the instantaneous design rate of heat gain is

$$\frac{q}{A} = 0.41\,(93 - 80) + 0.44\,(0.41)\,(129.7 - 93)$$

= 11.9 Btu per hour for each square foot of sunlit surface.

Example 6. Find the instantaneous design rate of heat gain through a 12-in. stone east wall ($b = 0.5; f_o = 4$) located in Lincoln, Nebr., at 5:00 p. m. sun time, when the temperature of the indoor air is constant at 80 F.

From Table 13, $U = 0.55$, the time lag is 8 hr, and $\lambda = 0.19$.

By linear interpolation on the basis of the value of b/f_o in Table 12,

$$t_m = 94.4 + \frac{0.125}{0.25}(105.9 - 94.4) = 100.2 \text{ F.}$$

Also from Table 12, the design sol-air temperature in Lincoln, Nebr., at a time earlier than 5:00 p. m. by the 8-hr time lag (at 9 a. m.) for an east wall is:

$$t'_e = 93 + \frac{0.125}{0.25}(142 - 93) = 117.5 \text{ F.}$$

From Equation 5, the instantaneous design rate of heat gain is:

$$\frac{q}{A} = 0.55\,(100.2 - 80) + 0.19\,(0.55)\,(117.5 - 100.2)$$

= 12.9 Btu per hour for each square foot of sunlit surface.

The time of the maximum instantaneous rate of heat gain from a sunlit wall or roof is later than the time of maximum rate of heat entry (maximum sol-air temperature) into the weather side by the time lag of the material. The times of maximum rate of heat entry into the weather

Table 14. Approximate Time of Maximum Rate of Heat Entry into Weather Side of Walls or Roofs in New York, N. Y., and in Lincoln, Nebr.[a,b]

Surface	Horizontal roof	North wall	East wall	South wall	West wall
Sun Time	12 noon to 1 00 p.m.	6.00 p.m.	9.00 a.m.	1:00 p.m.	4.00 p.m.

[a](Mid-summer; solar absorptivities of weather surface of 0.4 or greater.)

[b]For a completely shaded surface, the time of maximum rate of heat entry into the weather side is the time of maximum temperature of the outdoor air, which is at approximately 3:00 p. m., sun time, in both localities.

sides of sunlit walls or roofs having solar absorptivities greater than 0.4 are given in Table 14 for New York, N. Y., and for Lincoln, Nebr.

Example 7. Find the time of maximum instantaneous rate of heat gain from a south wall of 8 in. of common brick in New York, N. Y., if the wall is: (a) sunlit; (b) completely shaded.

(a) From Table 13, the time lag is 5.5 hr. From Table 14, the time of maximum rate of heat entry into the weather side is 1:00 p. m. The time of maximum instantaneous rate of heat gain is 6:30 p. m., sun time.

(b) The time of maximum instantaneous rate of heat gain from the completely shaded wall is 5.5 hr later than the time of maximum temperature of outdoor air (which is at 3:00 p. m.) or at 8:30 p. m., sun time.

Composite Walls or Roofs

A composite wall or roof is made up of two or more layers of different materials. For such a construction, the time lag is always slightly greater than the sum of the time lags of the individual layers. For example, in Table 13, assume that the time lag for a 12-in. wall of common brick were to be estimated by adding the time lags of the 8-in. brick wall and the 4-in. brick wall; the sum of these time lags is (5.5 + 2.3) or 7.8 hr, while the actual time lag of a 12-in. brick wall is 8.5 hr, or 0.7 hr greater than the sum. The value of the factor λ to be used in Equation 5 for a composite construction is less than the product of the λ values of the individual layers. Again, if the value of λ for a 12-in. west brick wall were to be estimated by using the products of the λ values for an 8-in. and a 4-in. brick wall, the result would be 0.44 (0.78) or 0.34; actually, the correct value of λ for a 12-in. west brick wall is 0.21.

Although an equation has been derived for the instantaneous rate of heat gain resulting from periodic heat flow through composite walls or roofs to an enclosure held at constant temperature, this equation is too complex for general use. The problem is receiving further study, and the methods of solution given here are approximate. It has been found, for example, that the *order* of the materials affects the time lag and the instantaneous rate of heat gain; a wall made of 2 in. of concrete and 1 in. of corkboard has a calculated time lag of 2 hr when the concrete is used for the outside layer and a time lag of 3 hr when the cork is used for the outside layer. Other things being the same, the use of the material of lower density as the weather-side layer will increase the time lag and decrease the instantaneous maximum rate of heat gain.

Two examples are given to show how to estimate, roughly, the instantaneous rate of heat gain through sunlit composite walls or roofs.

Example 8. Estimate the maximum instantaneous design rate of heat gain from a horizontal roof in New York, N. Y., that is made up of black built-up roofing on the weather side ($b = 1; f_o = 4$; thermal resistance = 0.28), 1 in. of insulating board, and 4 in. of concrete with no ceiling, when the temperature of the indoor air is 80 F.

The over-all coefficient of heat transfer for this construction is:

$$U = \frac{1}{0.25 + 0.28 + 3.03 + 0.33 + 0.61} = 0.22 \text{ Btu per (hour) (square foot) (Fahrenheit degree).}$$

If the time lag of the built-up roofing be ignored, the sum of the time lags of the individual layers is, from Table 13, (0.23 + 2.5) or 2.73 hr.

Actually, the time lag will be between 0.5 hr and 1.0 hr greater than this, so assume a time lag of 3.5 hr.

From Table 13, the homogeneous concrete roof having a time lag of 3.5 hr would have a value of λ of about 0.65; use this value for the composite roof.

With values of t_m and t'_e found from Table 11, as in previous examples, use Equation 5 and find the maximum design instantaneous rate of heat gain as:

$$\frac{q}{A} = 0.22\,(106.4 - 80) + 0.65\,(0.22)\,(155 - 106.4)$$
$$= 13 \text{ Btu per (hour) (square foot).}$$

The maximum instantaneous rate of heat gain from this roof would occur at about 3:30 p. m., sun time.

Example 9. Estimate the maximum instantaneous design rate of heat gain from a south wall in Lincoln, Nebr., consisting of 4 in. of face brick ($b = 0.7; f_o = 4$), 4 in. of common brick, furred, with an air space (thermal resistance = 0.75), and finished on the inside with ¾ in. of plaster on metal lath (thermal resistance = 0.23); the temperature of the indoor air is constant at 80 F.

The over-all coefficient of heat transfer for this construction is:

$$U = \frac{1}{0.25 + 0.44 + 0.80 + 0.75 + 0.23 + 0.61} = 0.32 \text{ Btu per (hour) (square foot)}$$
(Fahrenheit degree).

From Table 13, the sum of the time lags for the face brick and the common brick is (2.4 + 2.3) or 4.7 hr. The actual time lag will be slightly greater than this, and a value of 5.5 hr will be assumed.

From Table 13, a south wall of homogeneous common brick having a time lag of 5.5 hr will have a value of λ of 0.49; this value will be used for the composite wall.

By interpolation in Table 12, $t_m = 99.8$ F and $t'_e = 123.6$ F.

From Equation 5, the maximum instantaneous design rate of heat gain is:

$$\frac{q}{A} = 0.32\,(99.8 - 80) + 0.49\,(0.32)\,(123.6 - 99.8).$$
$$= 10.1 \text{ Btu per (hour) (square foot).}$$

The time of this heat gain is about 6:30 p. m., sun time.

Graphical Illustrations of Heat Flow-Time Relationship

Since sol-air temperature data are available for only two localities and since the time lag data for various wall constructions have not been completed, Figs. 1 through 8 are given to assist the designer in approximating cooling load requirements. The values given in Fig. 1 are for combined direct solar and sky radiation, and are given to represent the

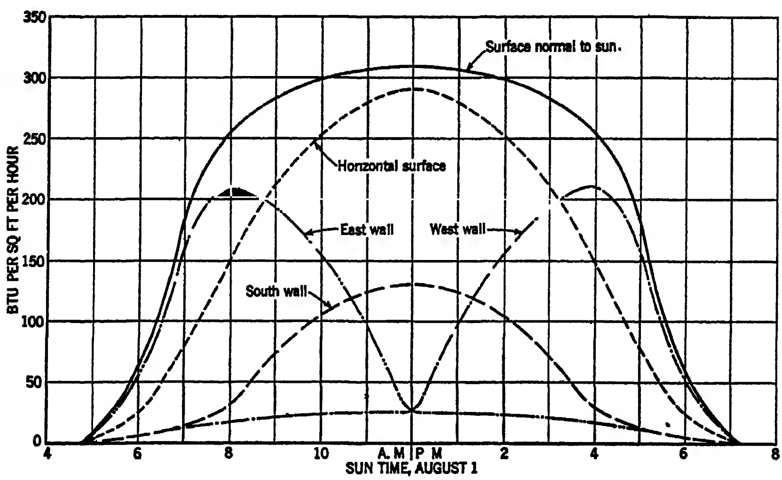

FIG. 1. SOLAR INTENSITY NORMAL TO SUN ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1 AT 40 DEG NORTH LATITUDE

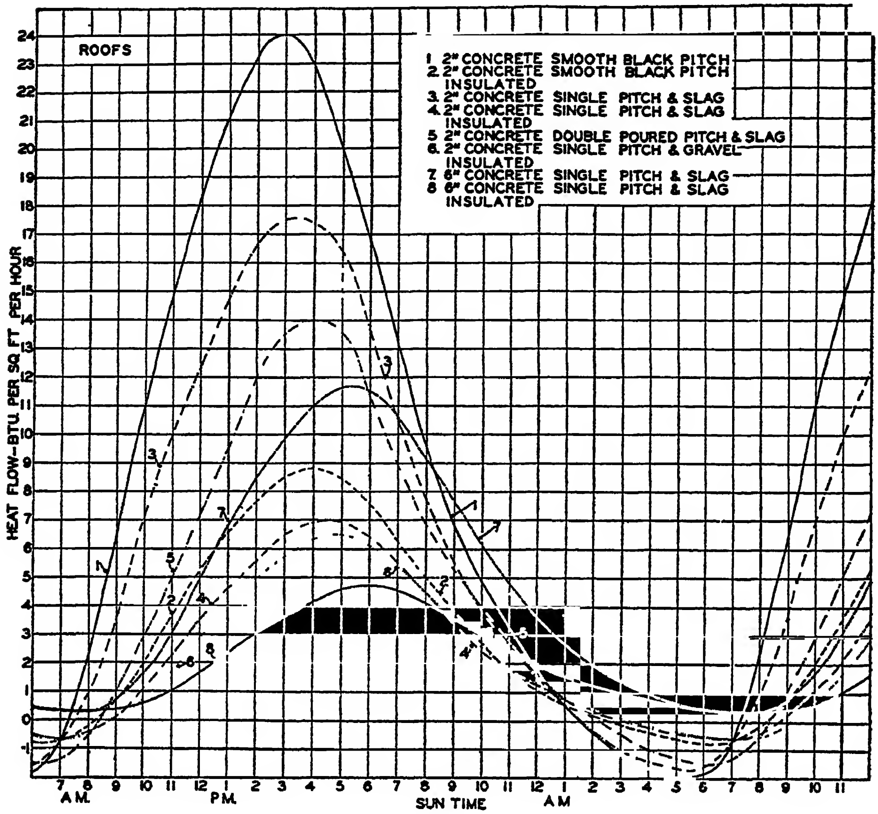

FIG. 2. HEAT FLOW-TIME RELATIONSHIP FOR HORIZONTAL ROOFS

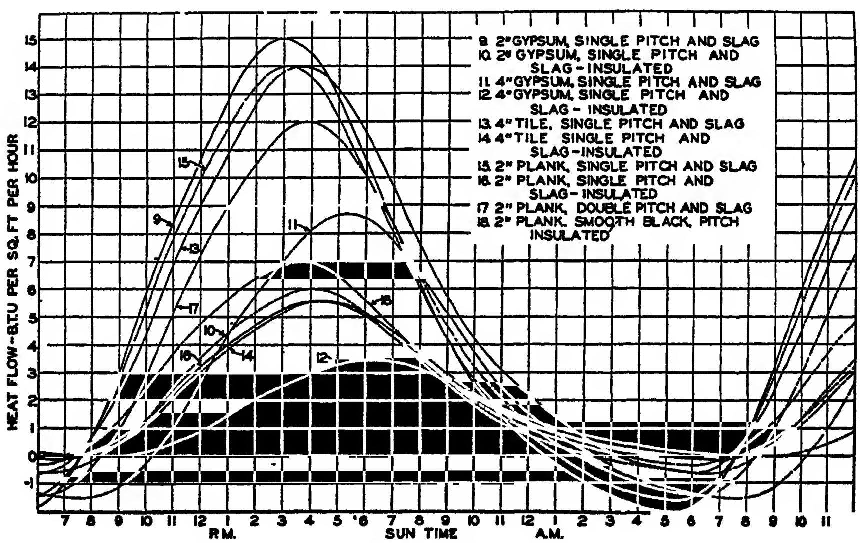

FIG. 3. HEAT FLOW-TIME RELATIONSHIP FOR HORIZONTAL ROOFS

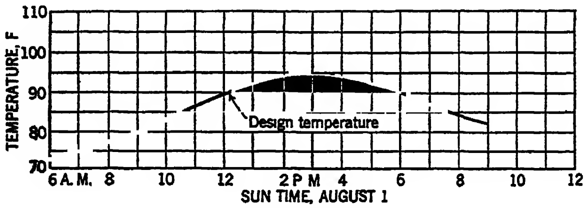

FIG. 4. OUTSIDE DESIGN TEMPERATURE BASIS FOR FIGS. 2 AND 3

expected design radiation intensity for August 1 for Pittsburgh, Pa. These curves were prepared by the A.S.H.V.E. Research Laboratory from data [7] obtained by pyrheliometer observations in Pittsburgh.

Figs. 2, 3, 5, 6, 7, and 8 illustrate the periodic relationship, variations in intensity, phase relation, and overlapping of solar intensities for variously oriented surfaces. Figs. 2 through 8 indicate the heat flow through the inside surfaces [8, 12] of the types of construction shown. Some authorities believe that the atmospheric conditions often encountered at Pittsburgh may have influenced these curves and that greater intensities may be encountered in other localities. Due to test deviations from design temperature, solar conditions, and typical construction, it is believed that the values indicated may be as much as 20 to 30 per cent below normal design expectations and can not be directly correlated with the data given for New York, N. Y., and Lincoln, Nebr.

The heat gain through southeast and southwest walls can be considered approximately the same as that through east and west walls respectively. The gain through northeast and northwest walls can be estimated as less than one half that through east and west walls respectively.

The time lags indicated in Table 15, based on tests in Pittsburgh, do not correspond to those in Table 13 which are based on a more recent source [8, 9] but relative values from Table 15 can be used to extend the application of Table 13 to constructions other than those shown in Table 13.

TABLE 15. TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

Type and Thickness of Wall or Roof	Time Lag, Hours
1-in. yellow pine horizontal roof, water proofing, smooth black finish	1
2-in. yellow pine horizontal roof, water proofing, smooth black finish	1¾
4-in. reinforced clay tile horizontal roof, water proofing, slag finish	2¼
2-in. gypsum horizontal roof, water proofing, slag finish	2¼
Slate and slaters felt on 2¼ in. tongue and grooved yellow pine, sloped roof	2½
4-in. gypsum horizontal roof, water proofing, slag finish	4½
6-in. concrete horizontal roof, water proofing, slag finish	5
1-in. concrete, 4-in. cinders, 1½-in. concrete, water proofing, smooth black finish	8
Wood siding, 1-in. sheathing, 2 x 4 studs, lath and plaster	2
Wood siding, 1-in. sheathing, 2 x 4 studs (studding space filled with insulation) lath and plaster	5
4-in. brick, 1-in. sheathing, 2 x 4 studs, lath and plaster	7
4-in. brick, 8-in. tile and plaster	10½
13-in. brick, plastered	12
9-in. brick, 3¾-in. tile, 5¼-in. air space, 3¾-in. tile and 1¼-in. plaster	16

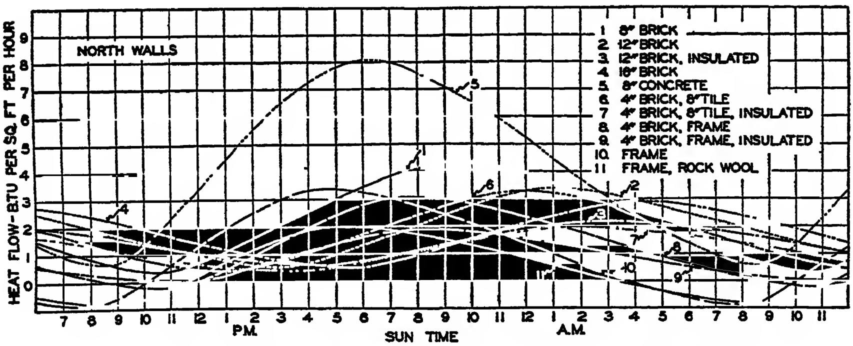

FIG. 5. HEAT FLOW-TIME RELATIONSHIP FOR NORTHERN EXPOSED WALLS

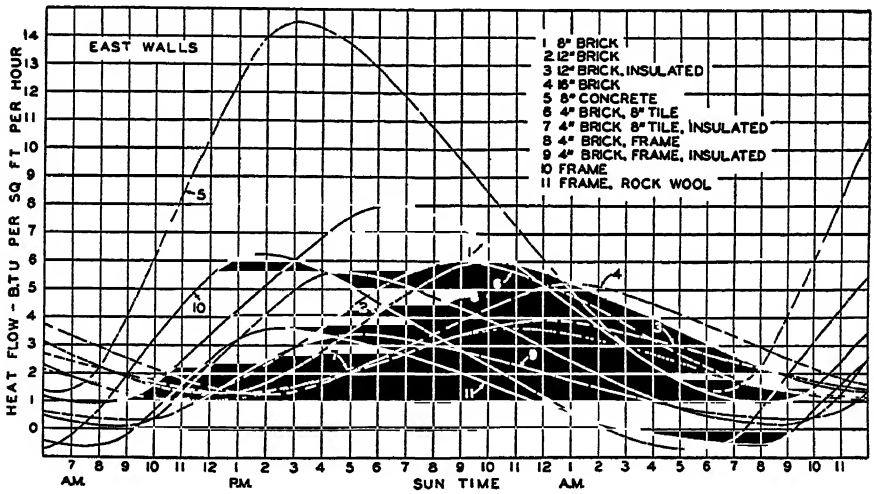

FIG. 6. HEAT FLOW-TIME RELATIONSHIP FOR EASTERN EXPOSED WALLS

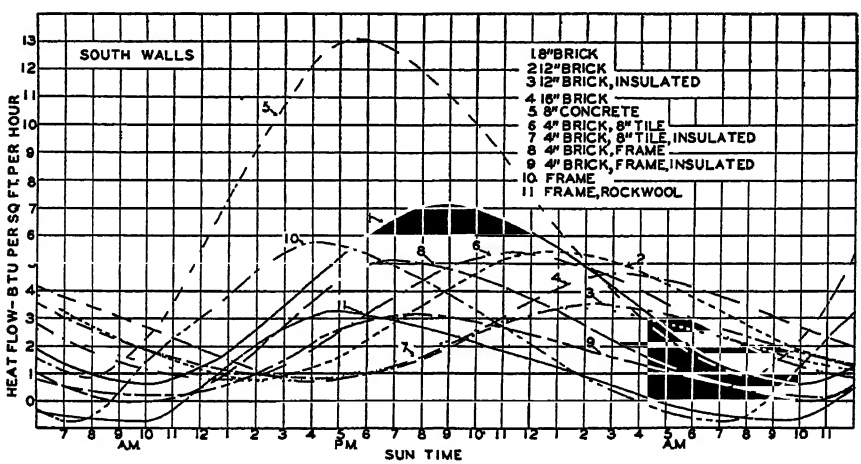

FIG. 7. HEAT FLOW-TIME RELATIONSHIP FOR SOUTHERN EXPOSED WALLS

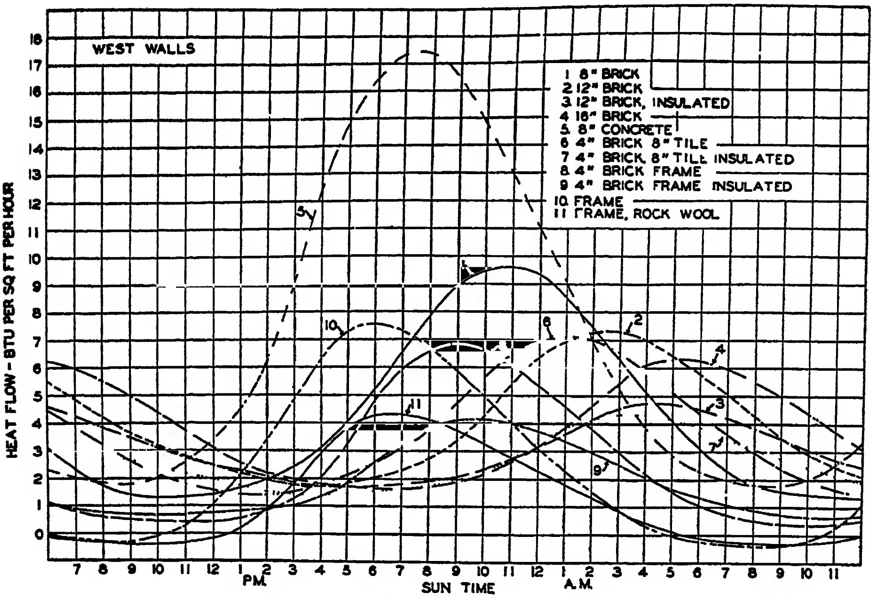

FIG. 8. HEAT FLOW-TIME RELATIONSHIP FOR WESTERN EXPOSED WALLS

HEAT EMISSION OF OCCUPANTS

The heat and moisture given off by human beings under different states of activity are shown in various tables and figures of Chapter 12 which covers the physical and physiological principles of air conditioning. It will be observed from these data that the rate of sensible and latent heat emission by human beings varies greatly depending upon state of activity. In many applications this component becomes a large percentage of total load. Metabolic rates are markedly variable for some extreme environmental conditions and this is another important factor which must be considered in cooling load computations. Consideration should be given to the typical age and sex of the occupants (whether men, women, or children) and the duration of the occupancy (since for short occupancy applications the extra heat and moisture brought in by people may be an important factor in the load).

HEAT GAIN FROM OUTSIDE AIR

When the temperature of the outdoor air is higher than the temperature of the indoor air, introduction of outdoor air for the purpose of ventilation or infiltration of outdoor air through cracks, openings, or doors, together with the exhaust or escape of room air, will result in a gain of sensible heat. This gain of heat is an immediate and undiminished source of cooling load. When the humidity ratio of the outdoor air exceeds the indoor humidity ratio, a gain of water vapor results from supply and exhaust or infiltration and escape of air; this gain becomes a latent heat load when heat must be removed by the cooling apparatus to condense the water vapor.

The purpose of supplying outdoor air and of exhausting room air is, generally, to reduce by dilution the concentration of odors and smoke.

In general, air for ventilation is supplied at the rate of 5 to 30 cfm per person; normal good practice is 10 to 20 cfm per person. Required or desirable rates of ventilation will be found in Chapter 12. The rate of entry of air due to infiltration may be estimated from data given in Chapters 8 and 14.

When the indoor air pressure reaches an equilibrium value (which is not greatly different from the outdoor air pressure), the rate of flow of room air from the structure must balance the rate of entry of outdoor air. Whether the air enters by infiltration through room openings or passes through the fan and other air conditioning apparatus is immaterial in this balance. Room air may be exhausted by use of exhaust fans or it may escape through room openings. It may be necessary to supply as much as 2000 or 3000 cfm of outside air through the apparatus to insure that air will escape rather than enter by infiltration through a single swinging door. Wind action and the location of openings relative to the neutral zone in the building are variables which make the estimate of quantity of outdoor air, necessary to insure outward leakage through selected openings, very approximate at best and subject to considerable judgment.

In the estimate of the amount of outside air which must be drawn through the apparatus, it is normally possible to subtract the assumed rate of infiltration from the required rate for ventilation.

The total rate of heat gain (combined sensible and latent) or the total contribution to the cooling load resulting from the introduction of outdoor air and the exhaust of room air is:

$$q_t = \frac{Q}{v_o} (h_o - h_i) \tag{6}$$

where

q_t = Quantity of heat (sensible and latent) to be removed from outdoor air, Btu per hour.
Q = Rate of entry of outdoor air, cubic foot per hour.
v_o = Volume of outdoor air per pound of dry air, cubic foot.
h_o = Enthalpy of outdoor air, Btu per pound of dry air.
h_i = Enthalpy of indoor air, Btu per pound of dry air.

Equation 6 includes the sensible heat gain and the latent heat gain. The actual gain of water vapor resulting from supply of outdoor air and exhaust of room air is:

$$M = \frac{Q}{v_o} (W_o - W_i) \tag{7}$$

where

M = Rate of gain of water vapor, pounds per hour.
W_o = Humidity ratio of outdoor air, pounds per pound dry air.
W_i = Humidity ratio of indoor air, pounds per pound dry air.

An example of the use of Equations 6 and 7 is given.

Example 10. For outdoor design conditions of 95 F dry-bulb and 75 F wet-bulb and indoor design conditions of 80 F dry-bulb and 67 F wet-bulb and for the supply of outdoor air at the rate of 1000 cfm and the exhaust of room air at the same weight rate, calculate: (a) the gain of water vapor, in pounds per hour; (b) the total gain of heat (both sensible and latent) in Btu per hour.

(a) From psychrometric data, $h_o = 38.26$, $h_i = 31.50$, $W_o = 0.01410$, $W_i = 0.01121$, and $v_o = 14.29$.

From Equation 7, $M = \frac{60\ (1000)}{14.29} (0.01410 - 0.01121) = 12.1$ pounds per hour.

TABLE 16. RATE OF HEAT GAIN FROM APPLIANCES **Without Hoods**[a,b]

Appliance	Capacity	Over-all Dimensions (Less Legs and Handles; Last Dimension is Height) Inches	Control A—Automatic M—Manual	Miscellaneous Data	Manufacturer's Rating	Maintaining Rate Btu per Hour	Recommended Rate of Heat Gain Btu per Hour		
							Sensible	Latent	Total
Restaurant Electrical Appliances									
Coffee Brewer and Warmer	½ gal		M M	Brewer 660 w Warmer 90 w	600[c] 90[c]	 306	900 230	220 60	1120 290
Coffee Brewer Unit with Tank	½ gal	20 x 30 x 26		2000 w Water heater, 2960 w brewer	4960[c]		4800	1200	6000
Coffee Urn	3 gal 5 gal	12 x 23 x 21 18 (Diam.) x 37	A A	Nickel plated Nickel plated	4500[c] 5000[c]	2600 3600	2200 3400	1500 2300	3700 5700
Doughnut Machine		22 x 22 x 57	A	Exhaust System	4700[c]		5000	0	5000
Egg Boiler	2 cups	10 x 13 x 25	M		1100[c]		1200	800	2000
Food Warmer, with Plate Warmer, per sq ft of top surface			A	Insulated, separate heat unit for each pot; plate warmer in base	400[c]	500	350	350	700
Food Warmer, alone, per sq ft of top surface			A		300[c]	400	200	350	550
Fry Kettle	11½ lb fat	12 (Diam.) x 14	A		2600[c]	1100	1600	2400	4000
Fry Kettle	25 lb fat	16 x 18 x 12	A	Area 12 x 14 in.	7000[c]	2000	3800	5700	9500
Griddle, Frying		18 x 18 x 8	A	Area 18 x 14 in.	2350[c]	2800	3100	1700	4800
Griddle, Frying		24 x 20 x 10	A	Area 23 x 18 in.	4000[c]	5000	5300	2900	8200
Grill, Meat		14 x 14 x 10	A	Area 10 x 12 in.	3000[c]	1900	3900	2100	6000
Grill, Sandwich		13 x 14 x 10	A	Area 12 x 12 in.	1650[c]	1900	2700	700	3400
Roll Warmer		23 x 23 x 29	A	Three drawers	1000[c]	900	2400	300	2700
Toaster, continuous	360 slices/hr	15 x 15 x 28	A	2 slices wide	2200[c]	5000	5100	1300	6400
Toaster, continuous	720 slices/hr	20 x 15 x 28	A	4 slices wide	3000[c]	6000	6100	2600	8700
Toaster, pop-up	216 slices/hr	12 x 11 x 9	A	4 slice	2450[c]	2000	4900	900	5800
Waffle Iron	20 waffles/hr	12 x 13 x 10	A	7 in. diam. waffle	750[c]	600	1100	750	1850

Restaurant Gas-Burning Appliances

Coffee Brewer and Warmer	½ gal		M M	Brewer Warmer	3400[d] 500[d]	500	1350 400	350 100	1700 500
Coffee Brewer Unit with Tank	4½ gal Tank	19 x 30 x 26		4 Brewers and Tank			7200	1800	9000
Coffee Urn	3 gal 5 gal	12 x 23 x 21 18 (Diam.) x 37	A A	Nickel plated Nickel plated		3400 4700	2500 3900	2500 3900	5000 7800
Food Warmer, per sq ft of top surface			M	Water bath	2000[d]	900	850	430	1280
Fry Kettle	15 lb fat	12 x 20 x 18	A	Area 10 x 10	14250[d]	3000	4200	2800	7000
Fry Kettle	28 lb fat	15 x 35 x 11	A	Area 11 x 16	24000[d]	4500	7200	4800	12000
Grill		22 x 14 x 17	M	Insulated, Grill surface of 1.4 sq ft Top burner 22,000 Btu/hr Bottom burner 15,000 Btu/hr	37000[d]		14400	3600	18000
Stoves, Short Order Open Top, per sq ft top Closed Top, per sq ft top Fry Top, per sq ft top			 M M M	 Ring type burners Ring type burners Tubular type burners	 14000[d] 11000[d] 12000[d]		 4200 3300 3600	 4200 3300 3600	 8400 6600 7200
Toaster, Continuous	360 slices/hr	15 x 15 x 28	A	2 slices wide	12000[d]	10000	7700	3300	11000
Toaster, Continuous	640 slices/hr	20 x 15 x 28	A	4 slices wide	20000[d]	14000	12000	5000	17000

Restaurant Steam-Heated Appliances

Coffee Urn	3 gal 5 gal	12 x 23 x 21 18 (Diam.) x 37	T T	Nickel plated Nickel plated			2400 3400	1600 2300	4000 5700
Coffee Urn	3 gal 5 gal	12 x 23 x 21 18 (Diam.) x 37	M M	Nickel plated Nickel plated			2600 3700	2600 3700	5200 7400
Food Warmer, per sq ft of top surface			T				400	500	900
Food Warmer, per sq ft of top surface			M				450	1150	1600

[a]For restaurant appliances, miscellaneous electrical and miscellaneous gas burning appliances.
[b]When these appliances are hooded and provided with adequate exhaust, use 50 per cent of recommended rate of heat gain from unhooded appliances.
[c]Manufacturer's rating in watts.
[d]Manufacturer's rating in Btu per hour.

TABLE 16. RATE OF HEAT GAIN FROM APPLIANCES Without Hoods[a,b] (CONCLUDED)

Appliance	Capacity	Over-all Dimensions (Less Legs and Handles; Last Dimension is Height) Inches	Control A—Automatic M—Manual	Miscellaneous Data	Manufacturer's Rating	Maintaining Rate Btu per Hour	Recommended Rate of Heat Gain Btu per Hour		
							Sensible	Latent	Total
Miscellaneous Electrical Appliances									
Hair Dryer, Blower Type			M	Fan, 165 w; Low, 915 w; High, 1580 w	1580[c]		2300	400	2700
Hair Dryer, Helmet Type			M	Fan, 80 w; Low, 300 w; High, 710 w	705[c]		1870	330	2200
Permanent Wave Machine			M	60 heaters at 25 w each, 36 in normal use	1500[c]		850	150	1000
Neon Sign, per linear ft of tube				½ in. outside diam. ⅜ in. outside diam.			30 60		30 60
Sterilizer, Instrument			A	For physicians; thermostat cuts off 550 w before boiling	1100[c]		650	1200	1850
Miscellaneous Gas-Burning Appliances									
Burners, Laboratory Small Bunsen Small Bunsen Fishtail Fishtail Large Bunsen		 7/16 in. Barrel 7/16 in. Barrel 7/16 in. Barrel 7/16 in. Barrel 1½ in. Mouth	 M M M M M	 Manufactured Gas Natural Gas Manufactured Gas Natural Gas Adjustable orifice	 1800[d] 3000[d] 3500[d] 5500[d] 6000[d]		 960 1680 1960 3080 3350	 240 420 490 770 850	 1200 2100 2450 3850 4200
Cigar Lighter			M	Continuous Flame	2500[d]		900	100	1000
Hair Dryer, 5 helmets			A	Heater and fan blowing air to helmets	33000[d]		15000	4000	19000
Stoves, Oven				Insulated, modern Not insulated	25000[d] 25000[d]	6000 8500	7200 9200	1800 2300	9000 11500

[a]For restaurant appliances, miscellaneous electrical and miscellaneous gas burning appliances.
[b]When these appliances are hooded and provided with adequate exhaust, use 50 per cent of recommended rate of heat gain from unhooded appliances.
[c]Manufacturer's rating in watts.
[d]Manufacturer's rating in Btu per hour.

(b) From Equation 6, the total rate of heat gain (combined sensible and latent) is:

$$q_t = \frac{60\ (1000)}{14.29}\ (38.26 - 31.50) = 28{,}400 \text{ Btu per hour.}$$

HEAT EMISSION OF APPLIANCES

Heat generating appliances which give off either sensible heat or both sensible and latent heat in an air conditioned enclosure may be divided into three general classes of equipment or devices: (1) electrical appliances, (2) gas appliances, and (3) steam heating appliances.

In the first group may be found such devices as lights[10], fans, motors, toasters, waffle irons, etc. The heat load caused by such devices may normally be obtained by multiplying the nameplate rating in watts by an appropriate load factor and by 3.4 (Btu per watt hour). In some cases it may also be possible to remove some of the heat of such appliances as lights and motors with exhaust air without involving it in the room load.

Electric motors are usually rated in units of horsepower *output*. To determine the corresponding input, which is the rate at which heat is added to the conditioned space by full-load operation of such motors, some idea of motor efficiency is necessary. The aggregate input in horsepower should then be multiplied by 2544 (Btu per horsepower hour).

Motor efficiencies can be assumed about as follows: Motor efficiencies vary from 50 to 60 per cent at the ⅛ hp level to 80 per cent at 1 hp and 88 per cent at 10 hp and above. Where the motor is outside of the conditioned space the heat equivalent of the motor output only is used, but where the motor is inside of the space the heat equivalent of the output divided by the efficiency is used.

In the second group belong such appliances as coffee urns, gas ranges, steam tables, broilers, hot plates, etc. For heat generating capacities of such appliances refer to Table 16 [13].

Judgment must be used in the application of data given in Table 16. Consideration must be given to the heat contributed by appliances which are in use at the time of peak load. The quantity of heat will depend upon whether products of combustion are vented to a flue, whether they escape into the space to be conditioned, or whether appliances are hooded allowing part of the heat to escape through a stack. There are no generally accepted data available on the effects of venting and shielding heating appliances but it is believed that, when they are properly hooded with a positive fan exhaust system through the hood, 50 per cent of the heat will be carried away and 50 per cent dissipated in the space to be conditioned. The same effectiveness of the hood should be figured for both latent and sensible heat.

MOISTURE THROUGH WALLS

In some applications walls of the conditioned space may be in contact with other spaces which have in them a higher water vapor pressure than that in the conditioned space. It is known that water vapor will flow through the building materials in proportion to the vapor pressure difference on the two sides of the material. The total amount of water vapor transmitted is dependent on the permeability which is usually expressed in grains of moisture per (square foot) (hour) (inch of mercury vapor pressure difference). The values for permeability in Table 17 are quoted from a publication of the *National Bureau of Standards*[11]. The

Table 17. Permeability of Various Materials to Water Vapor

Group	Material	Permeability Grains per (Sq Ft) (Hr) (Inch Hg)
1[a]	Plaster base and plaster, ¾ in.	14.7
	Fir sheathing, ¾ in.	2.9
	Waterproof paper[b]	49.1
	Pine lap siding	4.9
	Paint film	3.4
	Sugar cane fiberboard, ¾ in.	12.5
	Brick masonry, 4 in.	1.1
2[c]	Foil-surfaced reflective insulation, double-faced	0.08 to 0.13
	Roll roofing—smooth, 40 to 65 lb per roll 108 sq ft	0.13 to 0.17
	Duplex or laminated papers, 30-30-30	1.37 to 2.58
	Duplex or laminated papers, 30-60-30	0.52-0.86
	Duplex paper, coated with metallic oxides	0.52-1.29
	Insulation backup paper, treated	0.86-3 42
	Plaster, wood lath	11.00
	Plaster, 3 coats of lead and oil	3.68 to 3.84
	Plaster, 2 coats of Aluminum paint	1.15
	Plaster, fiberboard or gypsum lath	19.73 to 20.57
	Plywood, ½ in., 5-ply Douglas fir	2.67 to 2.74
	Plywood, 2 coats of asphalt paint	0.43
	Plywood, 2 coats of aluminum paint	1.29
	Gypsum Lath with metallic aluminum backing	0 09-0.39
	Insulating lath and sheathing, board type	25.68 to 34 27
	Insulating sheathing, surface-coated	3.03 to 4.36
	Insulating cork blocks, 1 in	6.19
	Mineral wool, unprotected, 4 in.	29 07
	Sheathing Paper, asphalt impregnated, glossy	0.17-2.05

[a]Calculating Vapor and Heat Transfer Through Walls, by L. G Miller (*Heating and Ventilating* 35, No. 11, 56, November, 1938).

[b]Light weight slaters felt used to keep rain from drifting through. Not used as a vapor barrier.

[c]How to Overcome Condensation in Building Walls and Attics, by L. V. Teesdale (*Heating and Ventilating*, Vol. 36, No. 4, April, 1939).

water vapor entering the conditioned space must be added to the latent cooling load.

Vapor barriers, to be effective in reducing entrance of moisture, must seal completely the walls, ceilings, and floors that are exposed to space having excessive vapor pressure and all doors must have gaskets applied to them to make the barrier effective.

EXAMPLE—COOLING LOAD ESTIMATE

From the foregoing discussion it is obvious that the determination of the maximum cooling load is rather complicated by reason of the variable nature of contributing load components. An illustrative example will explain the method presented in the foregoing text.

Example 11. A one-story office building is located in an eastern state near 40 deg latitude. The adjoining buildings on the north and west are not conditioned.

South wall construction: 8 in. concrete block, 4 in. brick veneer, ½ in. plaster on walls. (Table 8, Chapter 6, No. 92B, $U = 0.41$.) For summer, use $U = 0.40$.

East wall and outside north wall construction: 8 in. concrete block, ½ in. plaster on walls. (Table 7, Chapter 6, No. 82B, $U = 0.52$.) For summer, use $U = 0.50$.

West wall and adjoining north party wall construction: 13 in. solid brick, no plaster:

$$\frac{1}{U} = \frac{1}{1.65} + \frac{13}{5} + \frac{1}{1.65} \text{ or, } U = 0.263. \quad \text{Use } U = 0.26.$$

Roof construction: 2½ in. flat roof deck of gypsum fiber concrete on gypsum board. (Table 14, Chapter 6, No. 6A, $U = 0.38$.) For summer, use $U = 0.37$.

Floor construction: 4 in. concrete on ground.

Windows: 3 ft x 5 ft.

Front doors: Two 2 ft-6 in. x 7 ft (glass panels).

Side doors: Two 2 ft-6 in. x 7 ft (½ glass panels).

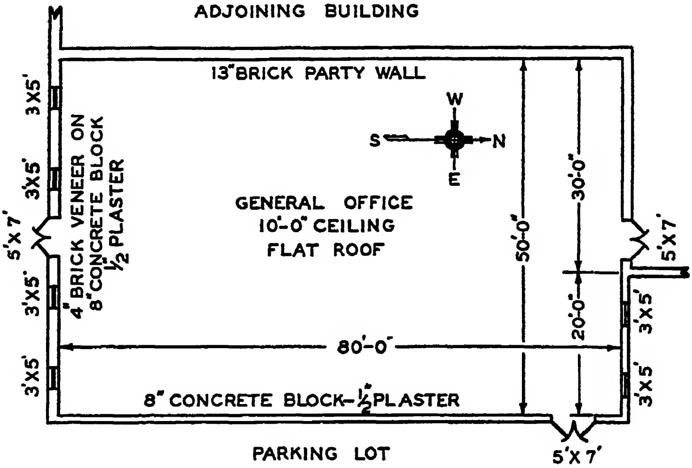

FIG. 9. PLAN OF ONE-STORY OFFICE BUILDING

Rear doors: Two 2 ft-6 in. x 7 ft (glass panels).

Occupancy: 85 office workers.

Lights: 16,000 watts.

Fan motor: 7½ hp.

Inside design conditions: dry-bulb 80 F; wet-bulb 65 F; W_1 = 0.0098 lb per pound dry air; h_1 = 29.94 Btu per pound dry air.

Outside design conditions: dry-bulb 95 F, wet-bulb 78 F.

Find: the total, sensible, and latent cooling loads.

Solution. Since the heat gain through the roof will constitute the major portion of the cooling load in this installation, the time lag of the roof should be found first in order to determine the approximate time of the maximum cooling load.

The roof consists of gypsum concrete and gypsum board; each of these materials has a density of between 50 and 60 lb per cubic foot. In Table 13, the roof material falls between wood and common brick on the basis of its density, which is an important property in periodic heat flow. For a thickness of 2½ in. of the roof deck, it seems reasonable to use a time lag of about 1.5 hr and a value of λ of 0.95. Sol-air temperature data for New York, N. Y., will be used as fairly representative of the location of the installation (see Table 11). Since the roofing is black, a value of b/f_o = 0.25 will be used. The maximum design sol-air temperature of the roof occurs at about 12 noon and is 155 F, (t'_e in Equation 5); the maximum contribution of the roof to the cooling load will occur about 1.5 hr later than the time of maximum sol-air temperature, or at about 1:30 p. m., sun time, and this will be assumed to be the time of the maximum cooling load.

From Table 11, t_m = 106.4 F. From Equation 5, the instantaneous rate of heat gain through the roof at 1:30 p. m., sun time, is

$$\frac{q}{A} = 0.37\ (106.4 - 80) + 0.95\ (0.37)\ (155 - 106.4) = 26.9 \text{ Btu per (hour) (square foot).}$$

The south wall consists of 4 in. brick veneer on 8 in. concrete block with plaster. Assume this wall is equivalent to 4 in. face brick on approximately 6 in. of solid concrete. From Table 13, the sum of the time lags of these materials is (2.4 + 3.8) or 6.2 hr; the actual time lag for a composite wall is slightly greater than this, and a time lag of 7 hr will be assumed for the south wall. Homogeneous south concrete with a time lag of 7 hr would have a value of λ of about 0.32. The design sol-air temperature for a south wall (b = 0.7; f_o = 4) at a time 7 hr earlier than 1:30 p. m. (at 6:30 a. m.) is t'_e = 79 F (Table 11 and Example 4); also from this table, t_m = 88.9 F. From Equation 5, the instantaneous rate of heat gain through the south wall at 1:30 p. m., sun time, is

$$\frac{q}{A} = 0.40\ (88.9 - 80) + 0.32\ (0.40)\ (79 - 88.9) = 2.3 \text{ Btu per (hour) (square foot).}$$

The exposed east wall and exposed north wall are constructed of 8 in. concrete block with ½ in. plaster; assume the hollow block to be equivalent to a solid 6 in. concrete

wall. From Table 13, the time lag of this wall would be about 3.8, say 4 hr. For the east wall, from Table 13, $\lambda = 0.46$; for the north wall, $\lambda = 0.61$. The design sol-air temperatures are found from Table 11 for 9:30 a. m. (4 hr earlier than the time of maximum cooling load); if the solar absorptivity of the weather side of the concrete be taken as 0.4 in Table 11:

For the east wall: $t'_e = 99$ F, $t_m = 87.8$ F.

For the north wall: $t'_e = 88.6$ F, $t_m = 86.3$ F.

From Equation 5, the instantaneous rate of heat gain through the east wall at 1:30 p. m., sun time, is:

$$\frac{q}{A} = 0.5\ (87.8 - 80) + 0.46\ (0.5)\ (99 - 87.8) = 6.8 \text{ Btu per (hour) (square foot).}$$

From Equation 5, the instantaneous rate of heat gain through the exposed north wall at 1:30 p. m., sun time, is:

$$\frac{q}{A} = 0.5\ (86.3 - 80) + 0.61\ (0.5)\ (88.6 - 86.3) = 3.9 \text{ Btu per (hour) (square foot).}$$

The north and west party walls, 13 in. brick, have a time lag (Table 13) of approximately 9 hr; also a reasonable average value of λ is 0.20. The temperatures in the unconditioned spaces will depend upon the amount of glass, orientation of the exposed surfaces, and amount of ventilation provided in the space. Assume the spaces to be ventilated during the night. The air temperature (t'_e) at 4:30 a. m., used in estimating the heat flow through these walls at 1:30 p. m., can be taken from Table 11 as 76 F. Assume the average temperature t_m as the design temperature of 95 F; then the rate of heat gain through these walls may be estimated from Equation 5 as:

$$\frac{q}{A} = 0.26\ (95 - 80) + 0.20\ (0.26)\ (76 - 95) = 2.9 \text{ Btu per (hour) (square foot).}$$

Since the floor is on the ground it is assumed to have no loss or gain.

A summary of the instantaneous rate of heat gain through walls and roof at 1:30 p. m., sun time, is given in the following table:

Surface	Net Area	q/A	Rate of Heat Gain
	Sq Ft	Btu/(hr) (sq ft)	Btu/hr
Roof	4000	26.9	107,600
South wall	405	2.3	930
East wall	765	6.8	5,200
Exposed north wall	170	3.9	660
West party wall	800	2.9	2,320
North party wall	265	2.9	770
Total instantaneous rate of sensible heat gain through walls and roof			117,480

The total area of glass in the south wall, including the doors, is 95 sq ft. To allow for the fact that the radiant energy transmitted by glass is absorbed principally by the floor and transferred after time lag to the air at a slower rate by convection, the cooling load at 1:30 p. m. through glass will be taken as 0.8 of the instantaneous rate of heat gain through south glass at 11:30 a. m.; from Table 6, and 40 deg north latitude, the instantaneous rate of heat gain through south glass at 11:30 a. m. is 89 Btu per (hour) (square foot); the sensible cooling load due to south glass is taken as $95 \times 0.8\ (89) = 6760$ Btu per hour.

The doors in the east wall have an area of 35 sq ft; the glass panel area will be taken as 18 sq ft and the heat gain through the solid panel will be ignored. By a calculation based upon Table 6 similar to the one given for south glass, the sensible cooling load due to east glass is: $18 \times 0.8\ (34.5) = 500$ Btu per hour.

There are windows on the exposed north side with an area of 30 sq ft. Since the instantaneous rate of heat gain through north glass during the middle of the day is very steady, the cooling load from north glass at 1:30 p. m. will be taken equal to the instantaneous rate of heat gain at this time, which, from Table 6, is: $30 \times 16 = 480$ Btu per hour.

The doors into the adjoining building on the north side have an area of 35 sq ft; time lag due to heat storage in the door is negligible, and the air temperature of the adjacent space will be taken as the design temperature of the outdoor air which is 95 F; rate of heat gain through north doors is: $35 \times 1.1 \times (95 - 80) = 580$ Btu per hour.

A summary of the sensible cooling load due to heat gain through windows and doors is given:

Surface	Btu per hr
South windows and doors	6760
East doors	500
North windows	480
North doors	580
Total sensible cooling load due to windows and doors	8320

The rate of sensible heat gain from lights is 16,000 × 3.4 = 54,400 Btu per hour.

The rate of sensible heat gain from the fan motor is: 7 5 × 2544 = 19,080 Btu per hour.

The rate of sensible heat gain from each moderately active office worker (metabolic rate of 500 Btu per hour) is 225 Btu per hour (Fig. 6, Chapter 12), so the rate of gain of sensible heat from occupants is: 85 × 225 = 19,120 Btu per hour.

Also by interpolation between Curves C and D of Fig. 7, Chapter 12, the rate of addition of water vapor to the room air from each moderately active office worker is 270 ÷ 1050 = 0.255 lb per hour, so the rate of gain of water vapor from occupants is: 85 × 0.255 = 21.68 lb per hour.

The design dry-bulb temperature of the outdoor air is 95 F, and the accompanying design wet-bulb temperature is 78 F. At standard atmospheric pressure, the design humidity ratio of the outdoor air is W_o = 0.0169 lb, the enthalpy is h_o = 41.46 Btu, and the volume is v_o = 14.37 cu ft. The minimum outdoor air required for air change or ventilation will be assumed to be 10 cfm for each occupant, or a total of 850 cfm, but in an attempt to prevent infiltration outdoor air will be supplied at the rate of 1200 cfm; since the volume of the conditioned space is 40,000 cu ft, the rate of air change is about 1.8 per hour.

From Equation 6, the total rate of heat gain (combined sensible and latent) from outdoor air is:

$$q_t = \frac{1200\ (60)}{14\ 37} (41.46 - 29.94) = 57{,}700 \text{ Btu per hour.}$$

From Equation 7, the gain of water vapor from outdoor air is at the rate of:

$$M = \frac{1200\ (60)}{14.37} (0.0169 - 0.0098) = 35.57 \text{ lb per hour.}$$

Approximate equivalent latent heat gain (based upon latent heat of condensation of 1050 Btu per pound, averaged at 75 F) is:

$$q_v = M \times h_{fg} = 35.57 \times 1050 = 37{,}350 \text{ Btu per hour.}$$

The rate of heat gain of sensible heat only from outside air is:

$$q_s = q_t - q_v = 57{,}700 - 37{,}350 = 20{,}350 \text{ Btu per hour.}$$

The gain of heat from outdoor air that passes through the cooling equipment does not affect the required state and weight rate of supply of conditioned air to the enclosure, but it does add to the load on the cooling equipment.

A summary of the estimated cooling load is given in Table 18.

Table 18. Summary of Estimated Cooling Load

Components of Sensible Cooling Load	Btu/Hr	Total Btu/Hr
Roof and Walls	117,480	
Windows and doors	8,320	
Lights	54,400	
Motor	19,080	
Occupants	19,120	
Outdoor air	20,350	238,750
Components of Latent Heat Gain[a]	**Btu/Hr**	
Occupants: 21 68 x 1050	22,750	
Outdoor air: 35.57 x 1050	37,350	60,100
Total Cooling Load on Cooling Apparatus		298,850

[a]Based upon latent heat of condensation of 1050 Btu per pound.

LETTER SYMBOLS USED IN CHAPTER 15

β = solar altitude, degrees.

γ = angle between the traces on a horizontal plane of the edge of the wall or frame providing shading, and the direct rays of the sun, degrees.

λ = a variable depending on thickness, material, and orientation of the wall or roof.

ρc = volumetric specific heat of building material, product of apparent density and gravimetric specific heat, Btu per (cubic foot) (Fahrenheit degree).

τ_s = transmissivity of glass for sky radiation (dimensionless).

τ_d = transmissivity of glass for direct solar radiation (dimensionless).

A = area of surface, square feet.

a_d = absorptivity of glass for direct solar radiation (dimensionless).

a_s = absorptivity of glass for sky radiation (dimensionless).

b = absorptivity of weather side of material for incident solar and sky radiation.

f_i = film coefficient of heat transfer of inside air, Btu per (square foot) (hour) (Fahrenheit degree).

f_o = film coefficient of heat transfer of outdoor air, Btu per (square foot) (hour) (Fahrenheit degree).

G_f = fraction of total window area which receives direct solar radiation.

h_{fg} = enthalpy of evaporation of water ($h_g - h_f$), Btu per pound.

h_i = enthalpy of indoor air per pound of dry air, Btu per pound.

h_o = enthalpy of outdoor air per pound of dry air, Btu per pound.

I = rate of incidence of solar and sky radiation, Btu per (square foot) (hour).

I_d = direct solar radiation incident upon glass, Btu per (square foot) (hour).

I_n = direct solar radiation at normal incidence, Btu per (square foot) (hour).

I_s = sky radiation incident upon glass, Btu per (square foot) (hour).

k = thermal conductivity of building material, Btu per (square foot) (hour) (Fahrenheit degree per foot)

l = height of window, feet.

L = thickness of building material, feet.

M = gain of water vapor from outdoor air supply and from exhaust of room air, pounds per hour.

q = instantaneous rate of heat gain from window, Btu per hour.

Q = rate of entry of outdoor air, cubic feet per hour.

q_s = $q_t - q_v$ = rate of gain of sensible heat only from outdoor air.

q_t = quantity of heat to be removed from outdoor air, Btu per hour.

q_v = approximate equivalent latent heat gain (based on latent heat of condensation of 1050 Btu per pound averaged at 75 F).

$\left(\frac{q}{A}\right) = \left(\frac{q}{A}\right)_1 + \left(\frac{q}{A}\right)_2$ = total instantaneous rate of heat gain from glass, Btu per (square foot) (hour).

$\left(\frac{q}{A}\right)_1$ = instantaneous rate of heat transfer from inside surface of glass (of heat conducted through glass and radiation absorbed by glass), Btu per (square foot) (hour).

$\left(\frac{q}{A}\right)_2$ = instantaneous rate of direct transmission of solar and sky radiation through glass, Btu per (square foot) (hour).

$\left(\frac{q}{A}\right)_L$ = instantaneous rate of heat entry into weather side of a sunlit building material that does not transmit directly solar or sky radiation, Btu per (square foot) (hour).

r_1 = s/l = ratio of window setback to window height.

r_2 = s/w = ratio of window setback to window width.

s = distance window is set back from plane of building wall.

t_a = outdoor air temperature, Fahrenheit degrees.

t_e = sol-air temperature, Fahrenheit degrees.

t'_e = sol-air temperature at a time earlier than the time for which heat gain is being found by an amount that is equal to the time lag of the wall or roof, Fahrenheit degrees.

t_i = indoor air temperature, Fahrenheit degrees.

t_L = temperature of weather surface of material, Fahrenheit degrees.

t_m = 24-hr average sol-air temperature for the particular value of $\frac{b}{f_o}$, Fahrenheit degrees.

U = over-all coefficient of heat transfer of wall or roof, Btu per (square foot) (hour) (Fahrenheit degree).

U_g = over-all coefficient of heat transfer of glass, Btu per (square foot) (hour) (Fahrenheit degree).

v_o = volume of outdoor air per pound of dry air, cubic feet.

w = width of window, feet.

W_i = humidity ratio of indoor air, pounds per pound of dry air.

W_o = humidity ratio of outdoor air, pounds per pound of dry air.

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CHAPTER 16

Fuels and Combustion

Classification of Coals, Cokes, Fuel Oils, and Gas, Dustless Treatment of Coal, Fundamental Principles of Combustion, Heat of Combustion, Air Required for Combustion, Excess Air, Heat Balance, Firing Methods, Secondary Air, Draft Requirements, Draft Regulation, Furnace Volume, Combustion of Gas, Soot, Condensation and Corrosion

FUELS may be classified according to their physical state as solid, liquid, or gaseous. The principal fuels used for domestic heating are coal, oil, and gas. However, coke, wood, kerosene, sawdust, briquettes, and other substances are used for heating in special applications or in localities where an adequate supply is available. Experiments are in progress in the use of a colloidal suspension of coal particles in fuel oil, but this fuel has not attained wide-spread usage as yet. The choice of fuel is usually based on dependability, cleanliness, availability, economy, operating requirements, and control.

CLASSIFICATION OF COALS

Coal has a complex composition that makes classification into clear-cut types difficult. Chemically it consists of carbon, hydrogen, oxygen, nitrogen, sulfur, and a mineral residue called ash. A chemical analysis provides some indication of the quality of a coal, but does not define its burning characteristics sufficiently. The coal user is interested principally in the available heat per pound of coal, in the handling and storing properties, the amount of ash and dust produced and the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space; a treatment applicable to heating boilers is given in a Bureau of Mines Bulletin .[1]

There are two forms of coal analysis; namely, the *proximate analysis* and the *ultimate analysis*. In the *proximate analysis* the proportions of moisture, volatile matter, fixed carbon, sulfur, and ash are determined. This analysis is more easily made and is satisfactory for indicating most of the characteristics which are of interest to the user. For the proximate analysis the moisture is determined by observing the loss of weight of a sample of coal when dried at about 220 F. To determine the volatile matter, the dried sample is heated to about 1750 F in a closed crucible, and the loss of weight is noted. The remaining sample is then burned in an open crucible, and the accompanying loss of weight represents the fixed carbon. The unburned residue is ash. Although determined separately, the sulfur content is frequently reported with the proximate analysis because the usefulness of a coal for certain purposes depends on its sulfur content.

In the *ultimate analysis*, which is difficult to make, the percentages of carbon, hydrogen, oxygen, nitrogen, sulfur, and ash in the coal sample are determined. It is used for detailed studies of fuels and in computing a heat balance when required in testing of heating devices. Typical ultimate analyses of the various kinds of coal are shown in Table 1. [2]

Other important qualities of coals are the screen sizes, ash fusion temperature, friability, caking tendency, and the qualities of the volatile matter. In considering these factors the following points are of interest.

TABLE 1. TYPICAL ULTIMATE ANALYSES FOR COALS

Rank	Btu per Lb		Constituents, Per Cent						
	Moist, Mineral-matter-free[a]	Moist, as Received	Oxygen	Hydrogen	Carbon	Nitrogen	Sulfur	Ash	O_2+H_2+C
Anthracite	14,600	12,910	5.0	2.9	80.0	0.9	0 7	10.5	87.9
Semi-Anthracite	15,200	13,770	5.0	3.9	80.4	1.1	1.1	8.5	89.3
Low-Volatile Bituminous	15,350	14,340	5.0	4.7	81.7	1.4	1.2	6.0	91.4
Medium-Volatile Bituminous	15,200	13,840	5.0	5.0	79.0	1.4	1.5	8.1	89.0
High-Volatile Bituminous *A*	14,500	13,090	9.2	5.3	73.2	1.5	2.0	8.8	87.7
High-Volatile Bituminous *B*	13,500	12,130	13.8	5.5	68.0	1 4	2.1	9.2	87.3
High-Volatile Bituminous *C*	12,000	10,750	21.0	5.8	60.6	1.1	2.1	9.4	87.4
Sub Bituminous *A*	......	......	......	......	......	......	......	......	......
Sub Bituminous *B*	10,250	9,150	29.5	6.2	52.5	1.0	1.0	9.8	88.2
Sub Bituminous *C*	9,000	8,940	35.8	6.5	46.7	0.8	0.6	9.6	89.0
Lignite	7,500	6,900	44.0	6.9	40.1	0.7	1.0	7.3	91.0

[a] $\left(\frac{100}{100 - 1\,1\ \text{Ash}}\right)$ (Btu as received).

The volatile products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 per cent of ash and moisture-free coal, increasing amounts of oils and tars are released. For coals of higher volatile content, the relative quantity of oils and tars decreases and is therefore low in the sub-bituminous coals and in lignite. The percentage of ash and its fusion temperature do not indicate the composition or distribution of its constituents.

A classification of coals is given in Table 2, and a brief description of the kinds of fuel is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other.

Anthracite is a clean, dense, hard coal which creates little dust in handling. It is comparatively hard to ignite but it burns freely when well started. It is non-caking, it burns uniformly and smokelessly with a short flame, and it requires no attention to the fuel bed between firings. It is capable of giving a high efficiency in the common types of hand-fired furnaces. A tabulation of the quality of the various anthracite sizes will be found in a Bureau of Mines Report [3]. Standard anthracite sizing specifications are shown in Table 3.

Semi-anthracite has a higher volatile content than anthracite. It is not so hard and ignites somewhat more easily; otherwise its properties are similar to those of anthracite.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling it. It ignites somewhat slowly and burns with a medium length of flame. Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak. Having only half the volatile matter content of the more abundant bituminous coals it can be burned with less production of smoke, and it is sometimes called *smokeless coal*.

The term *bituminous coal* covers a large range of coals and includes many types having distinctly different composition, properties, and burning characteristics. The coals range from the high-grade bituminous coals of the East to the poorer coals of the West Their caking properties range from coals which completely melt, to those from which the volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong and non-friable enough to

permit of the screened sizes being delivered free from fines. In general, they ignite easily and burn freely; the length of flame varies with different coals, but it is long. Much smoke and soot are possible, if improperly fired, especially at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly and have a medium length flame, are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value; it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking. The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed and pieces of char falling into the ashpit continue to burn. Very little smoke or soot is formed.

TABLE 2. CLASSIFICATION OF COALS BY RANK[a]

Legend: F.C. = Fixed Carbon. V M. = Volatile Matter. Btu = British thermal units.

Class	Group	Limits of Fixed Carbon or Btu Mineral-Matter-Free Basis	Requisite Physical Properties
I Anthracite	1. Meta-anthracite	Dry F.C., 98 per cent or more (Dry V.M., 2 per cent or less)	Non-agglomerating[b]
	2. Anthracite	Dry F.C., 92 per cent or more and less than 98 per cent (Dry V.M., 8 per cent or less and more than 2 per cent)	
	3. Semi-anthracite	Dry F.C., 86 per cent or more and less than 92 per cent (Dry V.M., 14 per cent or less and more than 8 per cent)	
II. Bituminous[d]	1. Low volatile bituminous coal	Dry F C., 78 per cent or more and less than 86 per cent (Dry V.M., 22 per cent or less and more than 14 per cent)	Either agglomerating[b] or non-weathering[f]
	2. Medium volatile bituminous coal	Dry F C., 69 per cent or more and less than 78 per cent (Dry V.M., 31 per cent or less and more than 22 per cent)	
	3. High volatile *A* bituminous coal	Dry F C, less than 69 per cent (Dry V.M., more than 31 per cent); and moist[c] Btu, 14,000[e] or more	
	4. High volatile *B* bituminous coal	Moist[c] Btu, 13,000 or more and less than 14,000[e]	
	5 High volatile *C* bituminous Coal	Moist Btu, 11,000 or more and less than 13,000[e]	
III. Sub-bituminous	1. Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[e]	Both weathering and non-agglomerating[b]
	2. Sub-bituminous *B* coal	Moist Btu, 9500 or more and less than 11,000[e]	
	3. Sub-bituminous *C* coal	Moist Btu, 8300 or more and less than 9500[e]	
IV. Lignitic	1. Lignite	Moist Btu less than 8300	Consolidated
	2 Brown coal	Moist Btu less than 8300	Unconsolidated

[a]This classification does not include a few coals which have unusual physical and chemical properties and which come within the limits of fixed carbon or Btu of the high-volatile bituminous and sub-bituminous ranks All of these coals either contain less than 48 per cent dry, mineral-matter-free fixed carbon or have more than 15,500 moist, mineral-matter-free Btu.

[b]If agglomerating, classify in low-volatile group of the bituminous class.

[c]Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

[d]It is recognized that there may be non-caking varieties in each group of the bituminous class.

[e]Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu.

[f]There are three varieties of coal in the high-volatile *C* bituminous coal group, namely, Variety 1, agglomerating and non-weathering; Variety 2, agglomerating and weathering; Variety 3, non-agglomerating and non-weathering.

Adapted from *A.S.T.M.* Standards, 1937, Supplement, p. 145, *American Society for Testing Materials* Philadelphia.

DUSTLESS TREATMENT OF COAL

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with various petroleum products, a solution of calcium chloride or a mixture of calcium and magnesium chlorides.

The coal is usually treated at the mine, but sometimes by the local distributor just before delivery. The salt solutions are sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size. Oil for the dustless treatment of coal is also applied under high pressure, in concentrations of 1 to 8 qt per ton of coal, depending upon the characteristics of the coal and oil.

Dustless treatments which are of such a corrosive nature that they may damage coal handling or burning equipment should not be used.

CLASSIFICATION OF COKES

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal or mixture of coals used, the temperatures and time of distillation and, to some extent, on the type of retort or oven; coke is also produced as a residue from the destructive distillation of oil.

High-temperature cokes. Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter. High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke,* which covers the greater part of the coke sold, and *gas-house coke.* The differences among these three cokes are relatively small; their denseness and hardness decrease and friability increases in the order named. In general, the lighter and more friable cokes ignite and burn the more easily.

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off. Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others briquetted.

Petroleum cokes, which are obtained by coking the residue left from the distillation of petroleum, vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

TABLE 3. STANDARD ANTHRACITE SIZING SPECIFICATIONS[a]

Size	Test Mesh, In.		Round Mesh			Maximum Impurities, Per Cent	
			Oversize	Undersize			
	Through	Over	Max. Per Cent	Max. Per Cent	Min. Per Cent	Slate[b]	Bone
Broken	4 3/8	3 1/4	——	15	7 1/2	1 1/2	2
Egg	3 1/4 to 3	2 7/16	5	15	7 1/2	1 1/2	2
Stove	2 7/16	1 5/8	7 1/2	12 1/2	7 1/2	2	3
Nut	1 5/8	13/16	7 1/2	10	5	3	4
Pea	13/16	9/16	10	15	7 1/2	4	5
Buckwheat	9/16	5/16	10	15	7 1/2	12 Ash	
Rice	5/16	3/16	10	15	7 1/2	13 Ash	
Barley	3/16	3/32	10	20	10	——	

[a]Approved and adopted by *Anthracite Committee,* State Street Building, Harrisburg, Pa.

[b]When slate content on Broken to Pea inclusive is less than above standards, bone content may be correspondingly increased, but slate content specified above shall not be exceeded in any event and the total maximum impurities shall not exceed those above specified.

TABLE 4. DETAILED REQUIREMENTS FOR FUEL OILS[a]

Grade[b]	Flash Point F Deg		Pour Point F Deg	Water and Sediment Per Cent	Carbon Residue Per Cent	Ash Per Cent	Distillation Temperatures F Deg				Viscosity Seconds			
							10 Per Cent Point	90 Per Cent Point		End Point	Saybolt Universal at 100 F		Saybolt Furol at 122 F	
	Min.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Min.	Max.	Max.	Min.	Max.	Min.
No. 1 Fuel oil—a distillate oil for use in burners requiring a volatile fuel.	100 or Legal	165	0[c]	Trace	0.05 on 10% Residuum[d]		410			560[e]				
No. 2 Fuel oil—a distillate oil for use in burners requiring a moderately volatile fuel.	110 or Legal	190	10[c]	0.05	0.25 on 10% Residuum[f]		440	600						
No. 3 Fuel oil—a distillate oil for use in burners requiring a low viscosity fuel.	110 or Legal	230	20[c]	0.10	0.15 Straight			675	600[g]		45			
No. 5 Fuel oil—an oil for use in burners requiring a medium viscosity fuel.	130 or Legal			1.00		0.10						50	40	
No. 6 Fuel oil—an oil for use in burners equipped with preheaters permitting a high viscosity fuel.	150			2.00[h]									300	45

[a]Recognizing the necessity for low sulfur fuel oils used in connection with heat-treatment, non-ferrous metal, glass and ceramic furnaces and other special uses, a sulfur requirement may be specified in accordance with the following table:

Grade of Fuel Oil	Sulfur, Max. Per cent
No. 1	0.5
No. 2	0 5
No. 3	0.75
No. 5	No Limit
No. 6	No Limit

Other sulfur limits may be specified only by mutual agreement between the buyer and seller

[b]It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c]Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0 F under any conditions.

[d]For use in other than sleeve type blue flame burners carbon residue on 10 per cent residuum may be increased to a maximum of 0 12 per cent. This limit may be specified by mutual agreement between the buyer and seller.

[e]The maximum end point may be increased to 590 F when used in burners other than sleeve type blue flame burners

[f]To meet certain burner requirements the carbon residue limit may be reduced to 0.15 per cent on 10 per cent residuum.

[g]The minimum distillation temperature of 600 F for 90 per cent may be waived if *A P I* gravity is 26 or lower

[h]Water by distillation, plus sediment by extraction Sum, maximum 2.0 per cent. The maximum sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

Table 5. Approximate Gravity and Calorific Value of Standard Grades of Fuel Oil

Commercial Standard No.	Approximate Gravity Range *A.P.I.*	Calorific Value Btu Per Gallon
1	38-40	136,000
2	34-36	138,500
3	28-32	141,000
5	18-22	148,500
6	14-16	152,000

CLASSIFICATION OF FUEL OILS

Fuel oils are produced by distillation from crude petroleum after gasoline, naphtha, and other lighter products have been removed. Fuel oil is composed chemically of about 85 per cent carbon and 12½ per cent hydrogen with small amounts of oxygen, nitrogen, and sulfur. Oils are classified according to their specific gravity, but specific gravity alone is not a sufficient index of the properties that are important for heating purposes. Other characteristics that must be considered in the choice of a fuel oil are the flash point, pour point, water and sediment content, carbon residue, ash, sulfur content, distillation temperatures, and viscosity.

The flash point and distillation characteristics are important relative to easy ignition and complete gasification of the oil in a burner. A low pour point and low water content are of interest in connection with the storage of the fuel in outdoor tanks, while a low viscosity permits easy passage through a small orifice. The sediment, carbon residue, and ash content should be low to prevent clogging of strainers and accumulation of unburned material in the burner. The sulfur content may be of importance because of the corrosive effect of sulfur compounds in the burner and heating appliance or in special commercial processes.

The Commercial Standard Specifications for Fuel Oils (CS 12-40) of the *U. S. Department of Commerce* are given in Table 4. These specifications conform to *American Society for Testing Materials* Tentative Specifications for Fuel Oils D 396-38T.

The relationship between the *A.P.I.* gravity of fuel oils and their calorific value is given in Table 5. Fuel oil grades No. 1, No. 2 and No. 3 only are used in domestic heating equipment. Grades No. 5 and No. 6 are used in commercial and industrial burners and usually require preheating.

CLASSIFICATION OF GAS

Gas is broadly classified as being either *natural* or *manufactured.* Natural gas is a mechanical mixture of several combustible and inert gases rather than a chemical compound. Manufactured gas as distributed is usually a combination of certain proportions of gases produced by two or more processes. Representative properties of gaseous fuels commonly used in domestic heating are presented in Table 6.

Natural gas is the richest of the gases and contains from 80 to 95 per cent methane, with small percentages of the other combustible hydrocarbons. In addition, it contains from 0.5 to 5.0 per cent of CO_2, and from 1 to 12 or 14 per cent of nitrogen. The heat value varies from 1000 to 1200 Btu per cubic foot, the majority of natural gases averaging about 1000 Btu per cubic foot. Table 6 shows typical values for the

Table 6. Representative Properties of Gaseous Fuels, Based on Gas at 60 F and 30 in. Hg.

Gas	Btu per Cu Ft		Specific Gravity, Air = 1.00	Air Required for Combustion, (Cu Ft)	Products of Combustion				Theoretical Flame Temperature, (F deg)
	High (Gross)	Low (Net)			Cubic Feet			Ultimate CO_2 Dry Basis	
					CO_2	H_2O	Total with N_2		
Natural gas—California	1200	1085	0.67	11.26	1.24	2.24	12.4	12.2	3610
Natural gas—Mid-Continental	970	870	0.57	9.17	0.97	1.92	10.2	11.7	3580
Natural gas—Ohio	1130	1025	0.65	10.70	1.17	2.16	11.8	12.1	3600
Natural gas—Pennsylvania	1130	1025	0.71	11.70	1.30	2.29	12.9	12.3	3620
Retort coal gas	570	510	0.42	5.00	0.50	1.21	5.7	11.2	3665
Coke oven gas	590	520	0.42	5.19	0.51	1.25	5.9	11.0	3660
Carbureted water gas	540	495	0.65	4.37	0.74	0.75	5.0	17.2	3815
Blue water gas	300	280	0.53	2.26	0.46	0.51	2.8	22.3	3800
Anthracite producer gas	135	125	0.85	1.05	0.33	0.19	1.9	19.0	3000
Bituminous producer gas	150	140	0.86	1.24	0.35	0.19	2.0	19.0	3160
Oil gas	575	510	0.35	4.91	0.47	1.21	5.6	10.7	3725

four main oil fields, although values from any one field vary materially.

Table 6 also gives the calorific values of the more common types of manufactured gas. Most states have legislation which controls the distribution of gas and fixes a minimum limit to its heat content. The gross or higher calorific value usually ranges between 520 and 545 Btu per cubic foot, with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different districts, nor at successive times in the same district. However, in any community the variations in gas composition are held within suitable limits so that the performance of approved gas appliances will not be adversely affected.

FUNDAMENTAL PRINCIPLES OF COMBUSTION

Combustion may be defined as the chemical combination of a substance with oxygen with a resultant evolution of heat. The rate of combustion depends partly upon the specific rate of reaction of the combustible substance with oxygen, partly upon the rate at which oxygen is supplied, and upon the temperature obtained due to surrounding conditions.

Complete combustion is obtained when all of the combustible elements in the fuel are oxidized with all of the oxygen with which they can combine. All of the oxygen supplied may not be utilized.

Perfect combustion is defined as the result of supplying the required

amount of oxygen for combination with all of the combustible elements of the fuel and utilizing all of the oxygen so supplied.

The oxygen required for the process of combustion is obtained from air which is a mechanical mixture of oxygen, nitrogen and small amounts of carbon dioxide, water vapor and inert gases. These inert gases are generally included with the nitrogen, and for engineering purposes the values given herewith may be used.

	By Volume, Per Cent	By Weight, Per Cent
Oxygen, O_2	20.9	23.15
Nitrogen, N_2	79.1	76.85

The combination of oxygen with the combustible elements and compounds of a fuel is in accordance with fixed laws. In the case of perfect combustion the reactions and resultant combinations are shown in Table 7.

The most important condition governing the process of combustion is temperature. It is necessary to bring a combustible substance to its ignition temperature before it will unite in chemical combination with oxygen to produce combustion. The ignition temperatures for several of the combustible constituents of fuels are presented in Table 7.

HEAT OF COMBUSTION

As previously stated, the process of combustion results in the evolution of heat. The heat generated by the complete combustion of a unit of fuel is constant for a given combination of combustible elements and compounds, and is known as the *heat of combustion*, *calorific value*, or *heating value* of the fuel. The heat of combustion of the several substances found in the more common fuels is given in Table 7.

The calorific value of a fuel may be determined either by direct measurement of the heat evolved during combustion in a calorimeter, or it may be computed from the ultimate analysis and the heat of combustion of the several chemical elements in the fuel. When the heating value of a fuel is determined in a calorimeter the water vapor is condensed and the latent heat of vaporization is included in the heating value of the fuel. The heating value so determined is termed the *gross* or *higher* heating value and this is what is ordinarily meant when the heating value of a fuel is specified. In burning the fuel, however, the products of combustion are not cooled to the dew-point and the higher heating value cannot be utilized.

When combustion is complete, the carbon in the fuel unites with oxygen to form carbon dioxide, CO_2, the hydrogen unites with oxygen to form water vapor, H_2O, and the nitrogen, being inert, passes through the reaction without change. When combustion is incomplete, some of the carbon may unite with oxygen to form carbon monoxide, CO, and some of the hydrogen and hydrocarbon gases may not be burned at all. When carbon monoxide or other combustible gases are present in the flue gases, considerably less heat is produced per unit of fuel consumed, and a lower combustion efficiency is obtained. Incomplete combustion may result from any or all of the following three conditions:

1. Inadequate air supply.
2. Insufficient mixing of the air and gases.
3. A temperature too low to produce ignition or maintain combustion.

TABLE 7. GENERAL DATA OF COMBUSTIBLE ELEMENTS AND COMPOUNDS [a]

Substance	Molecular Symbol	Chemical Reaction of Combustion	Ignition Temperature[a] F Deg	Calorific Value: Btu per Lb, Higher	Calorific Value: Btu per Lb, Lower	Calorific Value: Btu per Cu Ft[b], Higher	Theoretical Oxygen and Air Requirements: Lb per Lb, O_2	Lb per Lb, Air	Cu Ft per Cu Ft, O_2	Cu Ft per Cu Ft, Air
Carbon (to CO)	—	$2C + O_2 = 2CO$	—	3950	—	—	1.332	5.763	—	—
Carbon (to CO_2)	—	$2C + 2O_2 = 2CO_2$	—	14093	—	—	2.664	11.527	—	—
Sulfur (to SO_2)	—	$S + O_2 = SO_2$	—	3983	—	—	0 998	4.285	—	—
Sulfur (to SO_3)	—	$2S + 3O_2 = 2SO_3$	—	5940	—	—	1.497	6.428	—	—
Carbon Monoxide	CO	$2CO + O_2 = 2CO_2$	1166-1319	4347	—	321.8	0.571	2.471	0.5	2.382
Methane	CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	1260-1380	23879	21520	1013.2	3.990	17.265	2.0	9.528
Acetylene	C_2H_2	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	763-824	21500	20776	1499	3.073	13.297	2.5	11.911
Ethylene	C_2H_4	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	986-1123	21644	20295	1613.8	3.422	14.807	3.0	14.293
Ethane	C_2H_6	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	990-1120	22320	20432	1792	3.725	16.119	3.5	16.675
Hydrogen	H_2	$2H_2 + O_2 = 2H_2O$	1063-1166	60958[c]	51571[c]	325	7.937	34.344	0.5	2.382
Hydrogen Sulfide	H_2S	$2H_2S + 3O_2 = 2H_2O + 2SO_2$	599-608	7100	6545	647	1.409	6.097	1.5	7.146
Propane	C_3H_8	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	950-1080	21661	19944	2590	3.629	15.703	5.0	23.821
n-Butane	C_4H_{10}	$2C_4H_{10} + 13O_2 = 8CO_2 + 10H_2O$	890-1020	21308	19680	3370	3.579	15.487	6 5	30.967
Commercial Propane	—	—	—	21650	—	2500	—	—	—	—
Commercial Butane	—	—	—	21400	—	3200	—	—	—	—

[a] Values in table taken chiefly from page 51 of Fuel Flue Gases published by *American Gas Association.*
[b] Gas measured at 60 F and 30 in. Hg.
[c] Value from *National Bureau of Standards*

TABLE 8. APPROXIMATE THEORETICAL AIR REQUIREMENTS

SOLID FUEL	POUNDS AIR PER POUND FUEL
Anthracite	9.6
Semi-bituminous coal	11.2
Bituminous coal	10.3
Lignite	6.2
Coke	11.2

FUEL OIL	POUNDS AIR PER GALLON FUEL
Commercial Standard No. 1	102.6
Commercial Standard No. 2	104.5
Commercial Standard No. 3	106.5
Commercial Standard No. 5	112.0
Commercial Standard No. 6	114.2

GASEOUS FUELS	CUBIC FEET AIR PER CUBIC FOOT GAS
Natural gas	10.0
Mixed, natural and water gas	4.4
Carbureted water gas	4.4
Water gas, coke	2.1
Coke oven gas	5.2

AIR REQUIRED FOR COMBUSTION

The weight of air required for perfect combustion of a pound of fuel may be determined by use of the ultimate analysis of the fuel as applied to Equations 1 and 2. The various elements are expressed in percentages by weight.

Solid and Liquid Fuels:

$$\text{Pounds air required per pound fuel} = 34.56\left[\frac{C}{3} + \left(H - \frac{O}{8}\right) + \frac{S}{8}\right] \tag{1}$$

Gaseous Fuels:

$$\text{Pounds air required per pound fuel} = 2.46\,CO + 34.56\,H_2 + 17.28\,CH_4 + 13.29\,C_2H_2 + 14.81\,C_2H_4 + 16.13\,C_2H_6 + 6.10\,H_2S - 4.32\,O_2 \tag{2}$$

When the analysis is given on a volumetric basis the equation is expressed as follows:

$$\text{Cubic feet air required per cubic foot gas} = 2.39\,(CO + H_2) + 9.56\,CH_4 + 11.98\,C_2H_2 + 14.35\,C_2H_4 + 16.74\,C_2H_6 - 4.78\,O_2 \tag{3}$$

Equations 4 and 5 may be used as approximate methods of determining the theoretical air requirement for any fuel.

$$\text{Pounds air required per pound fuel} = 0.755 \times \frac{\text{Heating value (Btu per pound)}}{1000} \tag{4}$$

$$\text{Cubic feet air required per unit fuel} = \frac{\text{Heating value (Btu per unit)}}{100} \tag{5}$$

Approximate values for the theoretical air required for different fuels are given in Table 8.

It is customary to make use of the analysis of the products of combustion to determine the amount of flue gas produced and the actual amount of air supplied for combustion. The analysis of flue gases has been well described in various publications of the *Bureau of Mines* and

in the literature and the details of Orsat manipulation need not be considered in this discussion. (See Chapter 11.)

The weight of dry flue gas per pound of fuel burned is used in combustion loss calculations and may be determined by Equation 6.

$$\text{Pounds dry flue gas per pound fuel} = \frac{11\,CO_2 + 8\,O_2 + 7\,(CO + N_2)}{3\,(CO_2 + CO)} \times C \tag{6}$$

Values for CO_2, O_2, CO, and N_2 are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

EXCESS AIR

Since one measure of the efficiency of combustion is the relation existing between the amount of air theoretically required for *perfect* combustion and the amount of air actually supplied, a method of determining the latter factor is of value. Equation 7 will give reasonably accurate results, for most solid and liquid fuels, for determining the amount of air supplied per pound of fuel.

$$\text{Pounds dry air supplied per pound of fuel} = \frac{3.04\,N_2}{(CO_2 + CO)} \times C \tag{7}$$

Values for CO_2, CO, and N are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

The difference between the air actually supplied for combustion and the theoretical air required is known as excess air.

$$\text{Per cent excess air} = \frac{\text{Air supplied} - \text{Theoretical air}}{\text{Theoretical air}} \tag{8}$$

Since the calculation is usually made from Orsat analysis, Equation 9 will be found to be a convenient statement of this relationship.

$$\text{Per cent excess air} = \frac{100\left(O_2 - \frac{CO}{2}\right)}{N_2 \times 0.264 - \left(O_2 - \frac{CO}{2}\right)} \tag{9}$$

In this formula the symbols represent volumetric percentages of the flue gas constituents as determined by analysis.

Due to the different carbon-hydrogen ratios of the different fuels the maximum CO_2 attainable varies. Representative values for complete combustion of several fuels are given in Table 9.

Table 9. Representative Maximum CO_2 Value

Fuel	Theoretical CO_2	CO_2 Usually Attained in Practice
Coke	21.00	12-14
Anthracite	20.20	12-14
Bituminous Coal	18.20	13
No. 2 Fuel Oil	15.00	10.5
No. 6 Fuel Oil	16.50	13.5
Natural Gas	12.00	9.7
Coke Oven Gas	11.00	8.5

To produce heat efficiently with any of the common fuels the following requirements must be observed:

1. Adequate heat absorbing surface is necessary.
2. The heat transfer surfaces must be clean.
3. A minimum of excess air should be used.
4. The combustion air and the combustible gases produced by the fuel must be well mixed.
5. The quantity of combustible gases escaping to the stack must be kept small.

If insufficient heating surface is provided in a heating appliance, or if the heat transfer surfaces are covered with soot, ash or scale, the flue gas temperature will be excessive and the amount of sensible heat passing up the stack will be unnecessarily large. Too much excess air dilutes the flue gases excessively and increases the sensible flue gas loss, while a deficiency of air will cause some combustible gases to pass out of the appliance unburned. The highest combustion efficiency is not always obtained by supplying enough excess air to reduce the incomplete combustion loss to zero, but the incomplete combustion loss should be kept small. If the secondary air is not well mixed with the combustible gases, some incomplete combustion may still occur. Unnecessary secondary air also dilutes the flue gases and increases the sensible heat escaping up the chimney. Some excess air is always required in the practical operation of heating plants. It is considered good practice, under usual operating conditions, to supply from 25 to 50 per cent excess air, depending upon the fuel used.

HEAT BALANCE

In analyzing the performance of a heating appliance, it is frequently desirable to make an accounting, insofar as possible, of the disposition of all the heat units in the fuel used. Such an accounting is sometimes called a *heat balance*. The several components of the heat balance may either be expressed in terms of Btu per pound of fuel used or as a percentage of the calorific value of the fuel. The components of the heat balance are listed in items 1 to 7.

1. Useful heat transferred to heating medium and usually evaluated by determining the rate of flow of the heating fluid through the heating device and the change in enthalpy of the fluid (heat added) between the inlet and outlet.

2. Heat loss in the dry chimney gases.

$$h_1 = w_g c_p (t_g - t_a) \tag{10}$$

3. Heat loss in water vapor formed by the combustion of hydrogen.

$$h_2 = \frac{9H_2}{100} (1091.8 + 0.455\, t_g - t_a) \tag{11}$$

4. Heat loss in water vapor in the air supplied for combustion.

$$h_3 = 0.455\, M\, w_a (t_g - t_a) \tag{12}$$

5. Heat loss from incomplete combustion.

$$h_4 = 10143\, C \left(\frac{CO}{CO_2 + CO}\right) \tag{13}$$

6. Heat loss from unburned carbon in the ash or refuse.

$$h_5 = 14093 \left(\frac{C_u}{100} - C\right) \tag{14}$$

7. Radiation and all other unaccounted for losses.

Since the radiation and convection losses from a heating appliance are not usually determined by direct measurement, they, together with any other losses not measured, are determined by subtracting the total of items 1 to 6 inclusive from the heat of combustion of the fuel. Frequently, when there is CO in the flue gases there also will be small amounts of unburned hydrogen and hydrocarbon gases in the products of combustion. The loss represented by these unburned gases may easily be as large as that resulting from the presence of carbon monoxide. In this event item 7 of the heat balance would also include this unmeasured loss.

Symbols used in Equations 10 to 14 inclusive are:

h_1 = heat loss in the dry chimney gases, Btu per pound of fuel.
h_2 = heat loss in water vapor from combustion of hydrogen, Btu per pound of fuel.
h_3 = heat loss in water vapor in combustion air, Btu per pound of fuel.
h_4 = heat loss from incomplete combustion of carbon, Btu per pound of fuel.
h_5 = heat loss from unburned carbon in the ash, Btu per pound of fuel.
w_g = weight of dry flue gas per pound of fuel (from Equation 6), pounds.
c_p = mean specific heat of flue gases at constant pressure (c_p ranges from 0.242 to 0.254 for flue gas temperatures from 300 F to 1000 F)[2], Btu per pound.
t_g = temperature of flue gases at exit of heating device, Fahrenheit degrees.
t_a = temperature of combustion air, Fahrenheit degrees.
H_2 = percentage of hydrogen in fuel by weight from ultimate analysis of fuel as fired.
1091.8 = enthalpy of saturated water vapor at a temperature of 70 F, Btu per pound.
M = humidity ratio of combustion air, pounds of water vapor per pound of dry air.
w_a = weight of combustion air per pound of fuel used, pounds, from Equations 1, 2, 8, and 9.
C = weight of carbon burned per pound of fuel corrected for carbon in ash, pounds.

$$C = \frac{WC_u - W_a C_a}{100\ W} \tag{15}$$

where

C_u = percentage of carbon in the fuel by weight from the ultimate analysis.
CO, CO_2 = percentages of CO, CO_2 in flue gases by volume.
W_a = weight of ash and refuse, pounds.
C_a = per cent of combustible in ash by weight (combustible in ash is usually considered to be carbon).
W = weight of fuel used, pounds.

The flue gas losses listed as items 2, 3, and 4 of the heat balance may be determined with considerable accuracy from the curves shown in Fig. 1 [2] in many cases. The values of the losses plotted for fuel oil were computed from the ultimate analysis of a typical fuel oil used in domestic burners, while those plotted for the several ranks of coal were computed from the typical ultimate analyses shown in Table 1. The curves for medium volatile bituminous coal may be used for high volatile bituminous coal with negligible error. No curves are shown for gaseous fuel because various natural gases and manufactured gases vary considerably in their composition.

FIRING METHODS FOR ANTHRACITE

An anthracite fire should never be poked or disturbed, as this serves to bring ash to the surface of the fuel bed where it may melt into clinker.

Egg size is suitable for large fire-pots (grates 24 in. and over) if the fuel

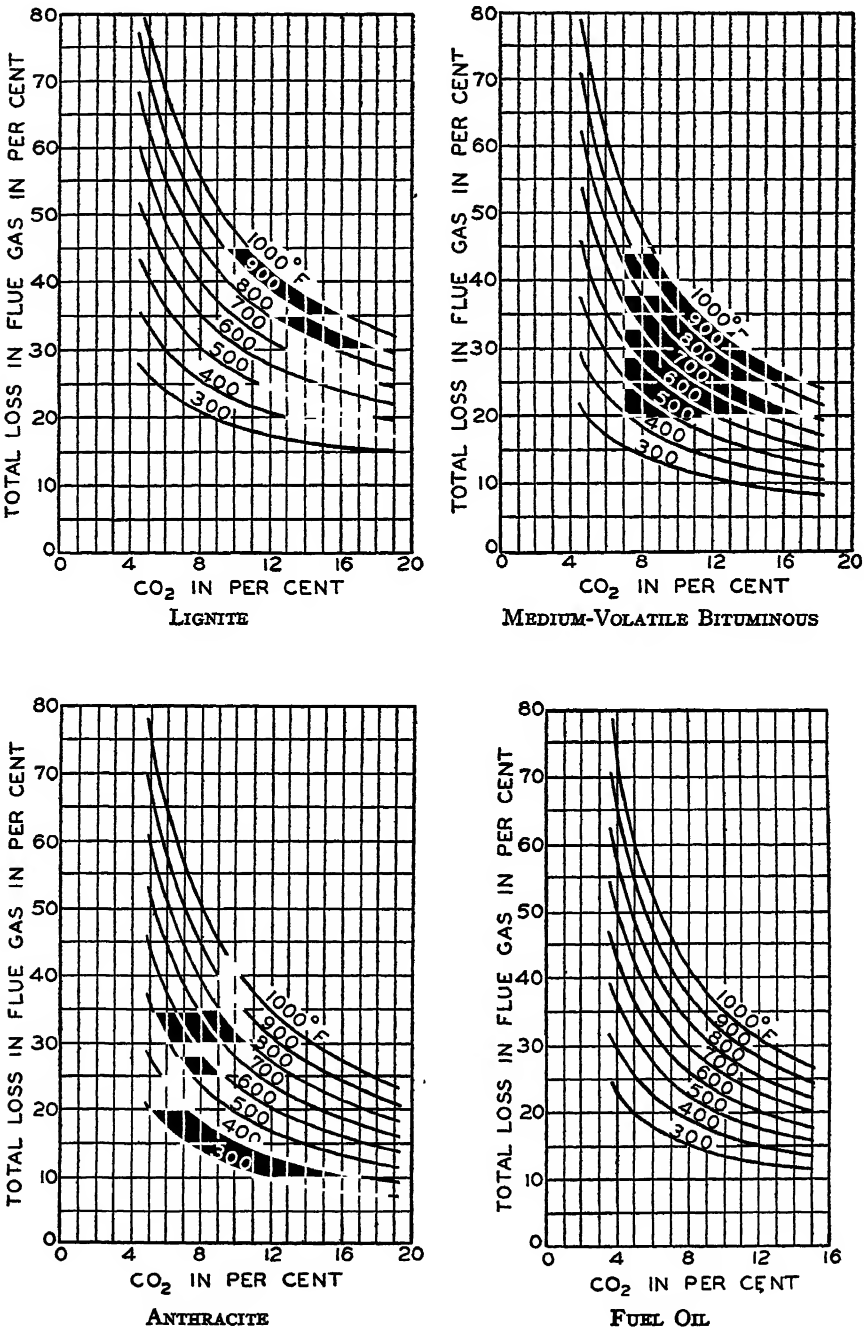

FIG. 1. FLUE GAS LOSSES WITH VARIOUS FUELS[2]

can be fired at least 16 in. deep. For best results this coal should be fired deeply.

Stove size coal is the proper size of anthracite for many boilers and furnaces used for heating buildings. It burns well on grates at least 16 in. in diameter and 12 in. deep. The fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for fire-pots up to 20 in. in diameter, with a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. When fired carefully, pea coal can be burned on standard grates. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire-door. A satisfactory method of firing pea coal consists of drawing the red coals toward the front end and piling fresh fuel toward the back of the fire-box.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open and regulating solely by means of the cold air check and the air inlet damper.

Buckwheat size coal for best results requires more attention than pea size coal, and in addition the smaller size of the fuel makes it more difficult to burn on ordinary grates. Greater care must be taken in shaking the grates than with the pea coal on account of the danger of the fuel falling through the grate. In house heating furnaces the coal should be fired lightly and more frequently than pea coal. When banking a buckwheat coal fire it is advisable after coaling to expose a small spot of hot fire by putting a straight poker down through the bed of fresh coal. This will serve to ignite the gas that will be distilled from the fresh coal and prevent delayed ignition within the fire-pot, which in some cases, depending upon the thickness of the bed of fresh coal, is severe enough to blow open the doors and dampers of the furnace. Where frequent attention can be given and care exercised in manipulation of the grates this fuel can be burned satisfactorily without the aid of any special equipment.

In general it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and consequently to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep the clinker in an easily broken up condition so that it readily can be shaken through the grate. Forced draft and small mesh grates are frequently used for burning buckwheat anthracite. For greater convenience, domestic stokers are used.

Buckwheat anthracite No. 2, or rice size, is used principally in stokers of the domestic, commercial and industrial type. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

FIRING METHODS FOR BITUMINOUS COAL

A commonly recommended procedure for firing domestic heating units, called the side-bank method, requires the movement of live coals to one side or the back of the grate, and placing the fresh fuel charge on the opposite side. The results are a more uniform release of volatile gases, and the subjection of these gases to the high temperature of the red coals. If the fresh charge is covered with a layer of fine coal, still better results may be obtained because of slower release of volatile matter.

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

The importance of firing bituminous coal in small quantities at short intervals is discussed in a U. S. Bureau of Mines technical paper[1]. Better combustion is obtained by this method in that the fuel supply is maintained more nearly proportional to the air supply.

If the coal is of the caking kind the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the fire-box. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

It is acknowledged that it may be difficult to apply the outlined methods to domestic heating boilers of small size, especially when frequent attendance is impractical. The adherence to these methods insofar as practical, however, will result in better combustion.

The output obtained from any heater with bituminous coal will usually exceed that obtained with anthracite, since bituminous coal burns more rapidly than anthracite and with less draft. Bituminous coal, however, will usually require frequent attention to the fuel bed.

Preventing Smoke

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Especial care must be taken in hand-firing bituminous coals.

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner fuel beds* on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher *A.P.I.* gravity oil always produces less smoke than a high volatile coal or low *A.P.I.* gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel burning equipment, or a change in the construction of the furnace, will often reduce

smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and a considerable investment in special apparatus is often necessary.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed will not as a general rule produce as much dust and cinders as will result from the burning of non-coking coals and slack coals when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of catcher must be installed.

FIRING METHODS FOR SEMI-BITUMINOUS COAL

The *Pocahontas Operators' Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone, the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring, and to gently rocking the grates. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

FIRING METHODS FOR COKE

Coke ignites less readily than bituminous coal and more readily than anthracite and burns rapidly with little draft. In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds rapidly to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly. In order to obtain the same interval of attention as with other fuels a deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather and should be shaken only until the first red particles drop from the grates in cold weather. The best size of coke for general use, for small fire-pots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in. screen. For large fire-pots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory. Large sizes of coke should be either mixed with fine sizes or broken up before using.

SECONDARY AIR

When bituminous coal is hand-fired in a furnace the volatile matter in the fuel distills off leaving coke on the grate. The product of combustion of the coke is CO_2 and under certain conditions some CO may arise from the bed. The combustion of the volatile matter and the CO may amount to the liberation of from 40 to 60 per cent of the heat in the fuel in the combustion space over the fuel bed.

The air that passes through the fuel bed is called *primary air* and the air that is admitted over the fuel bed in order to burn the volatile matter and CO is called *secondary air*.

This process of combustion is illustrated in Fig. 2 [4]. The free oxygen of the air passes through the grate and the ash above it and burns the carbon in the lower 3 or 4 in. of the fuel bed forming carbon dioxide. This layer noted as the oxidizing zone is indicated by the symbols CO_2 and O_2. Some of the carbon dioxide of the oxidizing zone is reduced to carbon monoxide in the upper layer of the fuel bed noted as the reducing zone and indicated by the symbols CO_2 and CO. The gases leaving the fuel

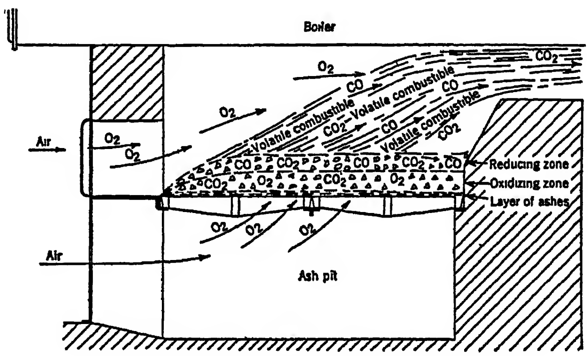

FIG. 2. COMBUSTION OF FUEL IN A HAND-FIRED FURNACE

bed are mainly carbon monoxide, carbon dioxide, nitrogen, and a small amount of free oxygen. Free oxygen is admitted through the firing door in an attempt to burn carbon monoxide and the volatile combustible distilled from the freshly fired fuel.

The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size and type of fuel, depth of fuel bed, and size of fire-pot.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not so important because small pieces fuse together and form large lumps. Fortunately a smaller size fuel gives more resistance to air flow through the fuel bed and thus automatically causes a larger draft above the fuel bed, which draws in more secondary air through the same slot openings, but, nevertheless, the smallest size of fuel will require the largest secondary air openings. For certain sizes of fuel no secondary air openings are required, and for large sizes, too much excess air may pass through the fuel bed.

In general, the efficiency of domestic hand-fired furnaces and boilers burning either anthracite or bituminous coal can be increased for an

hour or two after firing, if some secondary air is admitted through the slots of the fire door. However, unless the slots are closed when secondary air is no longer beneficial, the decrease in efficiency during the remainder of the firing cycle because of excess air may more than offset the gain resulting from the secondary air at the beginning of the firing period. Unless the secondary air can be readjusted between firings, it is probable that a greater average efficiency will be obtained for domestic hand-fired devices by leaving the secondary air slots closed at all times. There is usually an appreciable amount of air leakage around the firing door and secondary air slots of domestic furnaces and boilers.

When attention is given between firings the efficiency of combustion can be appreciably raised by admitting secondary air over a bituminous coal fire to burn the gases and reduce the smoke. The smoke produced is a good indicator, and that opening is best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial.

Secondary air that enters the combustion chamber too far removed from the zone of combustion will also be harmful, for the oxygen in the secondary air will not react with any unburned gases unless the mixture is subjected to high temperatures.

The air requirements of oil and gas burners are discussed in Chapter 17, Automatic Fuel Burning Equipment.

DRAFT REQUIREMENTS

The draft required to effect a given rate of burning the fuel is dependent on the following factors:

1. Kind and size of fuel.
2. Grate area.
3. Thickness of fuel bed.
4. Type and amount of ash and clinker accumulation.
5. Amount of excess air present in the gases.
6. Resistance offered by the boiler passes to the flow of the gases.
7. Accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel bed and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control that can be accomplished by adjustment of the dampers.

The quantity of excess air present has a marked effect on the draft required to produce a given rate of burning. If the excess is caused by holes in the fuel bed, or an extremely thin fuel bed, it is often possible to produce a higher rate of burning by increasing the thickness of the bed. The thickness of the fuel bed should not, however, be increased too much because the increased draft resistance will reduce the rate of primary air supply and the rate of burning.

For amount of draft required see Chapter 19, Chimneys and Draft Calculations.

DRAFT REGULATION

Because of the varying heating load demands present in most installations it is necessary to vary the rate of fuel burning. The maintenance of the proper air supply for the various rates of burning is accomplished

by regulation of the drafts. Correct and incorrect methods of draft regulation are shown in Fig. 3. The air enters through the ashpit draft door, firing door, and by leaks in the setting, whereas the gases leave only through the uptake. By throttling the gases with the damper in the uptake all the air entering by each of the three intakes is reduced in the same proportion, thus maintaining about the same per cent of excess air. If the ashpit draft door is closed, the air admitted through the ashpit is reduced, while it is increased through the other two intake openings, resulting in an increase of excess air. A considerable increase in the efficiency of hand-fired furnaces and boilers can be realized by regulating the air supply with the damper in the smokepipe instead of the ashpit damper. Use of the ashpit damper is required, of course, for low rates of combustion.

Methods of control of draft conditions when burning oil or gas are noted in Chapter 17, Automatic Fuel Burning Equipment.

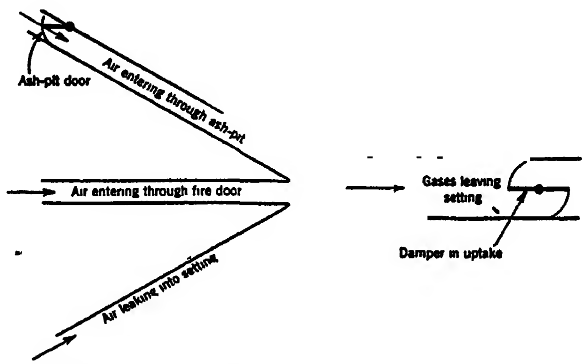

Fig. 3. Correct and Incorrect Methods of Draft Regulation in a Hand-Fired Furnace

FURNACE VOLUME

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and correctly proportioned combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals provision should be made for mixing the combustible gases thoroughly, so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly. Anthracite requires comparatively little combustion space.

COMBUSTION OF GAS

The majority of gas burners utilized in central domestic heating plants are of the Bunsen type and operate with a non-luminous flame. In this

type of burner part of the air required for combustion is mixed with the gas as primary air, the air and gas mixture being fed to the burner ports. Additional secondary air is introduced around the flame by draft inspiration. In the luminous flame burner, which is sometimes used, all of the air for combustion is brought in contact with the flame as secondary air. This secondary air should be brought into intimate contact with the gas.

Some makes of burners use radiants or refractories to convert some of the energy in the gas to radiant heat. The radiants also serve as baffles in directing the flow of the products of combustion.

The quantity of air given in Table 6 is that required for theoretical combustion, but with a properly designed and installed burner the excess air can be kept low. In order to insure freedom from carbon monoxide under conditions which may obtain in installations, it is customary to design gas burning appliances for a supply of 30 to 35 per cent of excess air. In individual installations in which flue gas analyses are made, the excess air is sometimes reduced to approximately 20 per cent. The division of the air into primary and secondary is a matter of burner design, the pressure of gas available, and the type of flame desired.

The air gas ratio has a decided effect upon flame propagation. It is necessary that the gas will flow out of the burner ports fast enough so that the flame cannot travel back into the burner head, i.e. *flash back*, but the velocity must not be so high that it blows the flame away from the port.

The maximum and minimum flow speeds from burner ports which may be permitted are known to be very close together when air-gas mixtures in theoretical proportions are being supplied to the burner. As the air-gas ratio is lowered, and the mixture becomes more *gas rich*, the limiting speeds become farther apart, until with 100 per cent gas, in an all-yellow flame, flash back cannot occur and a much higher velocity is needed to blow off the flames.

SOOT

The deposit of soot on the flue surfaces of a boiler or heater acts as an insulating layer over the surface and reduces the heat transmission to the water or air. The *Bureau of Mines Report of Investigations* No. 3272[5] shows that the loss of seasonal efficiency is not so great as has been believed and usually is not over 6 per cent because the greater part of the heat is transmitted through the combustion chamber surfaces. The *Bureau of Standards Report* BMS 54[6] points out that, although the decrease in efficiency of an oil fired boiler due to soot deposits is relatively small, the attendant increase in stack temperature may be considerable.

The soot accumulation clogs the flues, reduces the draft, and may prevent proper combustion. Soot can probably be most effectively removed by a jet of compressed air or by means of a brush. However, it has been found that copper chloride, lead chloride, tin chloride, zinc chloride, common salt and some other salts are partially effective in removing soot from furnaces and boilers when properly used.[7]

CONDENSATION AND CORROSION

Sulfur dioxide or sulfur trioxide formed by the combustion of sulfur in fuels is the principal corroding element in flue gases, and becomes active whenever moisture is present for the formation of sulfurous or sulfuric acid. It is necessary, therefore, to maintain a flue gas temperature in excess of the dew-point temperature of the flue gases in all parts of

TABLE 10. AVERAGE FLUE GAS DEW-POINT FOR VARIOUS FUELS[8]

TYPE OF FUEL	AVERAGE DEW-POINT TEMPERATURE, F
Anthracite	68
Semi-Bituminous Coal	84
Bituminous Coal	93
Oil	111
Natural Gas	127
Manufactured Gas	137

appliances unless they are made of materials that will resist these corrosive influences. It is usually desirable to maintain a flue gas temperature above the dew-point temperature throughout the heating appliance and the chimney or smokestack because of these same corrosive effects. The average dew-point temperatures of the flue gases from the several fuels, when burned with the amount of excess air usually supplied to insure complete combustion, are shown in Table 10.

LETTER SYMBOLS USED IN CHAPTER 16

h_1 = heat loss in the dry chimney gases, Btu per pound of fuel.
h_2 = heat loss in water vapor from combustion of hydrogen, Btu per pound of fuel.
h_3 = heat loss in water vapor in combustion air, Btu per pound of fuel.
h_4 = heat loss from incomplete combustion of carbon, Btu per pound of fuel.
h_5 = heat loss from unburned carbon in the ash, Btu per pound of fuel.
w_g = weight of dry flue gas per pound of fuel (from Equation 6), pounds.
c_p = mean specific heat of flue gases at constant pressure.
t_g = temperature of flue gases at exit of heating device, Fahrenheit degrees.
t_a = temperature of combustion air, Fahrenheit degrees.
H_2 = percentage of hydrogen in the fuel by weight from ultimate analysis of fuel as fired.
M = humidity ratio of combustion air, pounds of water vapor per pound of dry air.
w_a = weight of combustion air per pound of fuel used, pounds.
C = weight of carbon burned per pound of fuel corrected for carbon in ash, pounds.
C_u = percentage of carbon in the fuel by weight from the ultimate analysis.
CO, CO_2 = percentages of CO, CO_2 in the flue gases by volume.
W_a = weight of ash and refuse, pounds.
C_a = per cent of combustibles in ash and refuse by weight.
W = weight of fuel used, pounds.

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CHAPTER 17

Automatic Fuel Burning Equipment

Classification of Stokers, Combustion Process and Adjustments, Furnace Design, Classification of Oil Burners, Combustion Chamber Design, Classification of Gas-Fired Appliances, Combustion Process, Ratings, Fuel Burning Rates

AUTOMATIC mechanical equipment for the combustion of solid, liquid, and gaseous fuels is considered in this chapter.

MECHANICAL STOKERS

A mechanical stoker is a device that feeds a solid fuel into a combustion chamber, provides a supply of air for burning the fuel under automatic control and, in some cases, incorporates a means of removing the ash and refuse of combustion automatically. Coal can be burned more efficiently by a mechanical stoker than by hand firing because the stoker provides a uniform rate of fuel feed, better distribution in the fuel bed and positive control of the air supplied for combustion.

CLASSIFICATION OF STOKERS ACCORDING TO CAPACITY

Stokers may be classified according to their coal feeding rates. The following classification has been made by the *U. S. Department of Commerce*, in cooperation with the *Stoker Manufacturers Association.*

Class 1. Capacity under 61 lb of coal per hour.
Class 2. Capacity 61 to 100 lb of coal per hour.
Class 3. Capacity 101 to 300 lb of coal per hour.
Class 4. Capacity 300 to 1200 lb of coal per hour.
Class 5. Capacity 1200 lb of coal per hour and over.

Class 1 Stokers

These stokers are used primarily for home heating and are designed for quiet, automatic operation. Simple, trouble-free construction and attractive appearance are desirable characteristics of these small units.

A common stoker in this class (Fig. 1) consists essentially of a coal hopper, a screw for conveying the coal from the hopper to the retort, a fan which supplies the air for combustion, a transmission for driving the coal feed worm, and an electric motor for supplying power for coal feed and air supply.

Air for combustion is admitted to the fuel through tuyeres at the top of the retort which may be either round or rectangular. Stokers in this class are made for burning anthracite, bituminous, semi-bituminous, and lignite coals, and coke. The *U. S. Department of Commerce* has issued commercial standards for household anthracite stokers[1].

Units are available in either the hopper type, as shown in Fig. 1 or in the bin-feed type as shown in Figs. 2 and 3. Some stokers, particularly those designed for use with anthracite, automatically remove ash from the ash pit and deposit it in an ash receptacle as shown in Fig. 3. Most of the bituminous models, however, require removal of the ash from the fuel bed after it is fused into a clinker.

Stokers in this class feed coal to the furnace intermittently in accordance with temperature or pressure demands. A special control is used

to insure sufficient stoker operation to maintain a fire during periods when no heat is required. Where year around domestic hot water is supplied by a boiler and indirect water heater connected to a storage tank, the stoker will usually be called on to operate often enough to maintain the fire.

Stoker-Fired Boiler and Furnace Units

Boilers, air conditioners, and space heaters especially designed for stokers are available having design features closely coordinating the heat absorber and the stoker. Although efficient and satisfactory performance can be obtained from the application of stokers to existing boilers and furnaces, some of the combination stoker-fired units (Fig. 4) are more compact and attractive in appearance.

Class 2 and 3 Stokers

Stokers in this class are usually of the screw feed type without auxiliary plungers or other means of distributing the coal. They are used extensively for heating plants in apartments and hotels, also, for industrial

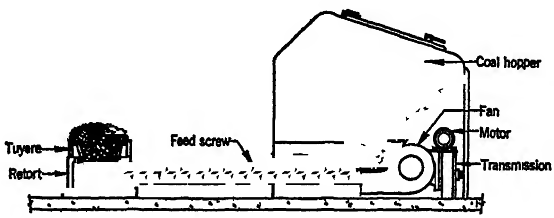

FIG. 1. UNDERFEED STOKER, HOPPER TYPE, CLASS 1

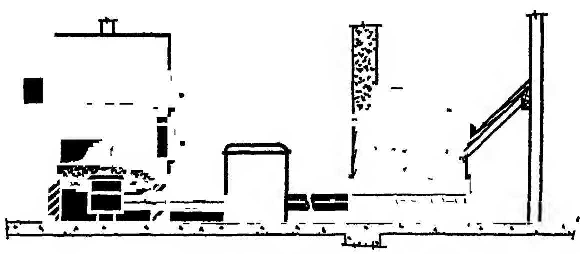

FIG. 2. UNDERFEED STOKER, BIN FEED TYPE, CLASS 1

plants. They are of the underfeed type and are available in both the hopper type, as illustrated in Fig. 5, and the bin feed type, shown in Fig. 6. These units also are built in plunger feed type with an electric motor or a steam or hydraulic cylinder coal feed drive.

Stokers in this class are available for burning all types of anthracite, bituminous and lignite coals. The tuyere and retort design varies according to the fuel and load conditions. Stationary type grates are used on bituminous models and the clinkers formed from the ash accumulate on the grates surrounding the retort.

Anthracite stokers in this class are equipped with moving grates which discharge the ash into a pit below the grate. This ash pit may be located on one or both sides of the grate and on some installations is of sufficient capacity to hold the ash for several weeks' operation.

Class 4 Stokers

Stokers in this group vary widely in details of design and several methods of feeding coal are employed. The underfeed stoker is widely used, although a number of the overfeed types are used in the larger sizes. Bin-feed, as well as hopper models, are available in both underfeed and overfeed types.

Class 5 Stokers

The prevalent stokers in this field are: (1) underfeed side cleaning, (2) underfeed rear cleaning, (3) overfeed flat grate, and (4) overfeed inclined grate.

Underfeed side cleaning stokers are made in sizes up to approximately 500 boiler horsepower. They are not so varied in design as those in the smaller classes, although the principle of operation is similar. A stoker of this type is illustrated in Fig. 7.

The rear cleaning underfeed stoker is usually of the multiple retort design and is used in some of the largest industrial plants and central power stations. Zoned air control has been applied to these stokers, both longitudinally and transversely of the grate surface.

The overfeed flat grate stoker is represented by the various chain—or traveling-grate stokers. A typical traveling-grate stoker is illustrated in Fig. 8.

Another distinct type of overfeed flat-grate stoker is the spreader (Figs. 9 and 10) type in which coal is distributed either by rotating paddles or by air over the entire grate surface. This type of stoker is adapted to a wide range of fuels and has a wide application on small sized fuels, and on fuels such as lignites, high-ash coals, and coke breeze.

The overfeed inclined-grate stoker operates on the same general combustion principle as the flat-grate stoker, the main difference being that rocking grates, set on an incline, are provided in the former to advance the fuel during combustion.

Combustion Process

In anthracite stokers of the Class 1 underfeed type, burning takes place entirely within the stoker retort. The refuse of combustion spills over the edge of the retort into an ash pit or receptacle from which it may be removed either manually or automatically.

Larger underfeed anthracite stokers operate on the same principle, except that the retort is rectangular and the refuse spills over only one or two sides of the grate. Anthracite for stoker firing is usually the No. 1 buckwheat or No. 2 buckwheat size.

Because the majority of the smaller bituminous coal stokers operate on the underfeed principle, a general description of their operation is given. When the coal is fed into the retort, it moves upward toward the zone of combustion and is heated by conduction and radiation from the burning fuel in the combustion zone. As the temperature of the coal rises, it gives off moisture and occluded gases, which are largely non-combustible. When the temperature increases to around 700 or 800 F the coal particles become plastic, the degree of plasticity varying with the type of coal.

A rapid evolution of the combustible volatile matter occurs during and directly after the plastic stage. The distillation of volatile matter continues above the plastic zone where the coal is coked. The strength and porosity of the coke formed will vary according to the size and characteristics of the coal. While some of the ash fuses into particles on the surface

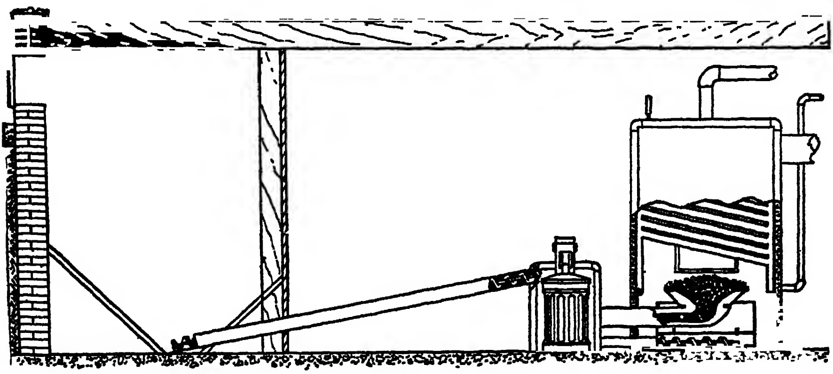

Fig. 3. Underfeed Anthracite Stoker with Automatic Ash Removal, Bin Type

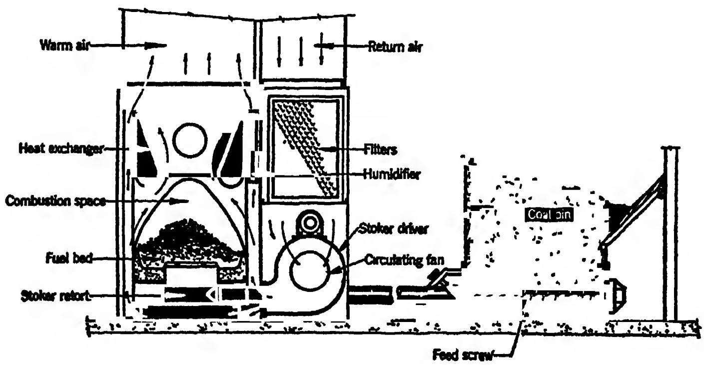

Fig. 4. Stoker-Fired Winter Air Conditioning Unit

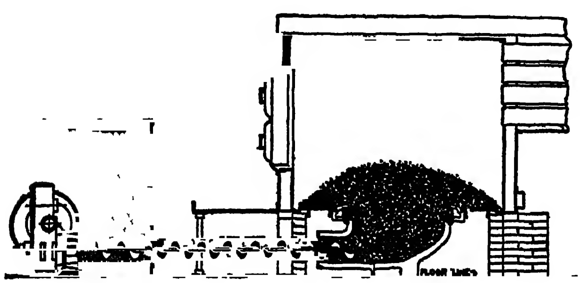

Fig. 5. Underfeed Screw Stoker, Hopper Type, Class 2, 3 or 4

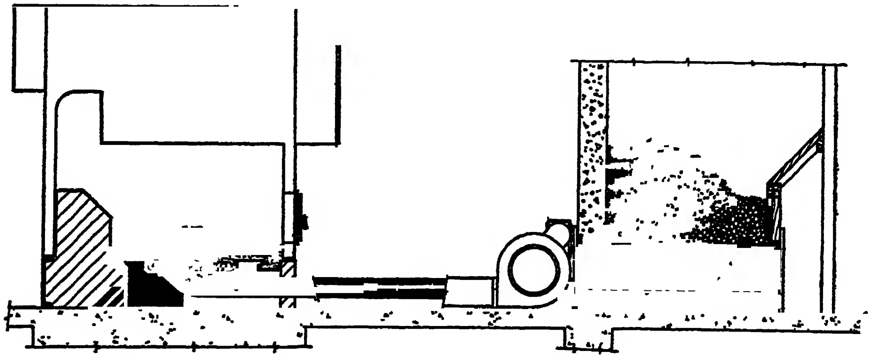

FIG. 6. UNDERFEED SCREW STOKER, BIN TYPE, CLASS 2, 3 OR 4

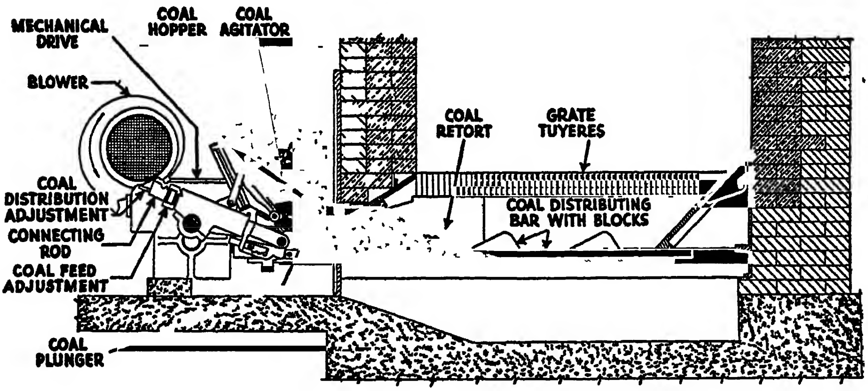

FIG. 7. UNDERFEED SIDE CLEANING STOKER

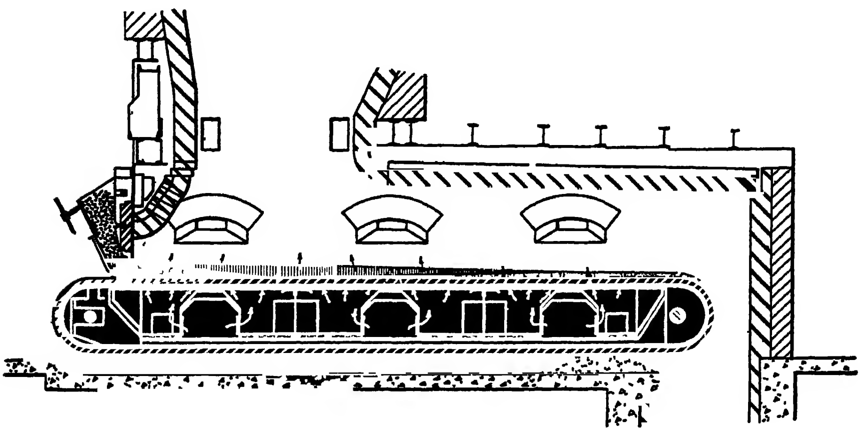

FIG. 8. OVERFEED TRAVELING-GRATE STOKER

of the coke as it is released, most of it remains on the hearth or grates and, as this ash layer becomes thicker with time, that portion exposed to the higher temperatures surrounding the retort fuses into a clinker. The temperature in the fuel bed, the chemical composition and homogeneity of the ash, and the time of heating govern the degree of fusion.

Most bituminous coal stokers of Classes 1, 2, 3 and 4 require manual removal of the ash in clinker form.

In the underfeed side-cleaning stokers the fuel is introduced at the front of the furnace to one or more retorts, and is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of stoker is suitable for all bituminous coals while in the smaller sizes it is suitable for small sizes of anthracite. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process the volatile gases are released, are mixed with air, and pass through the fire where they are burned. The ash may be continuously or periodically discharged at the sides.

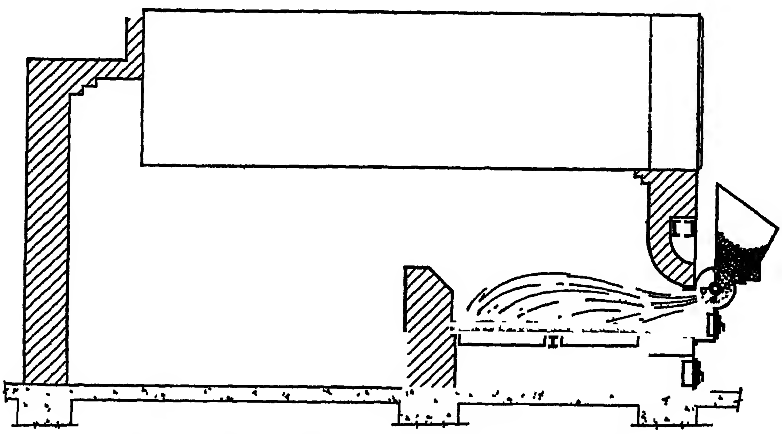

FIG. 9. OVERFEED SPREADER STOKER (ROTOR TYPE)

The underfeed rear-cleaning stoker accomplishes combustion in much the same manner as the side-cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side cleaning underfeed type.

Overfeed flat-grate stokers receive fuel at the front of the grate in a layer of uniform thickness and move it horizontally to the rear of the furnace. Air is supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ash pit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze, and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the fuel bed to maintain ignition of the incoming fuel and, frequently, a rear combustion arch.

In addition to the use of rocking grates, the overfeed inclined-grate stoker is provided with an ash plate on which ash is accumulated and

dumped periodically. This type of stoker is suitable for all types of coking fuels but preferably for those of low volatile content. Its grate action keeps the fuel bed broken up thereby allowing free passage of air. Because of its agitating effect on the fuel it is not desirable for badly clinkering coals. It usually should be provided with a front arch to ignite the volatile gases.

Combustion Adjustments

The coal feeding rate and air supply to the stoker should be regulated so as to maintain a balance between the load demand and the heat liberated by the fuel. Under such conditions no manual attention to the fuel bed should be required, other than the removal of clinker in stokers which operate on this principle of ash removal.

As in all combustion processes, the maintenance of the correct proportions of air and fuel is essential. It is desirable to supply the minimum amount of air required to properly burn the fuel at the rate of feed.

While there may be only slight variations in the rate at which the coal is being fed due to variations in the size or density of the coal, there may

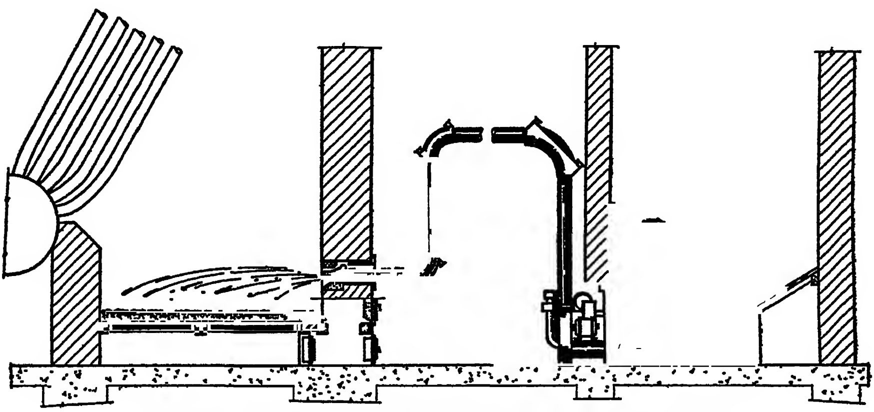

FIG. 10. OVERFEED SPREADER STOKER (PNEUMATIC TYPE)

be wide variations in the rate of air flow as the result of changes in fuel bed resistance. These changes in resistance may be caused by changes in the porosity of the fuel bed due to variations in size or friability of the coal, ash and clinker accumulation, and variations in depth of the fuel bed. Because of this variable fuel bed resistance, many bituminous stokers, even in the smaller domestic sizes, incorporate air controls which automatically compensate for these changes in resistance and maintain a constant air fuel ratio. The efficiency of combustion may be determined by analyzing the flue gases as explained in Chapter 16.

It is desirable on most stoker installations to provide automatic draft regulation in order to reduce air infiltration and provide better control during the banking, or off, periods of the stoker.

Furnace Design

Although there is considerable variation in stoker, boiler, and furnace design, the stoker industry, from long-time experience, has established certain rules for the proportioning of furnaces for domestic, and commercial stokers. The stoker installer and designer of stoker-fired equipment should give careful consideration to these factors.

The *Stoker Manufacturers Association* has published standard recommendations on setting heights for stokers having capacities up to 1200 lb of coal per hour[2].

The empirical formulas for determining these setting heights are:

For burning rates up to 100 lb coal per hour

$$H = 0.1125\,B + 15.75 \tag{1}$$

For burning rates from 100 to 1200 lb coal per hour

$$H = 0.03\,B + 24 \tag{2}$$

where

H = minimum setting height, inches, measured from dead plates to crown sheet for steel boilers. For cast-iron boilers height may be ⅞ H.

B = burning rate coal per hour, pounds.

Standards for minimum firebox dimensions and base heights have been formulated by the *Stoker Manufacturers Association* as shown in Fig. 11[2].

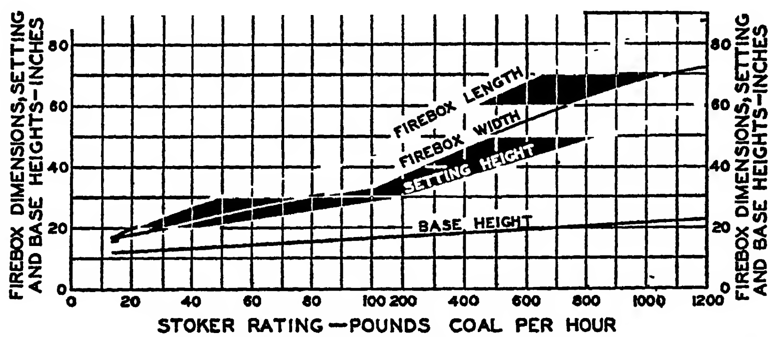

Fig. 11. Suggested Minimum Firebox Dimensions and Base Heights[a]

[a]For reference in selecting or designing boilers and furnaces for stoker firing. Dimensions shown are for net inside clearance at grate level using coal with heating value of not less than 12,000 Btu per pound. Under certain conditions smaller fireboxes will permit satisfactory performance but these dimensions are preferred normal minimums.

In considering these recommendations, it should be understood that they show the average recommended minimum. There are many factors affecting the proper application of stokers to various types of boilers and furnaces, and, in certain instances, setting height or firebox dimensions shown in the standards may be modified without impairing performance. Such modification rests with the experience of the installer, or designer, with a particular stoker, the type of fuel used, and the construction of the boiler or furnace.

Installation of stokers (particularly smaller sizes) from the side of the boiler or furnace will sometimes facilitate clinker removal.

Rating and Sizing Stokers

The capacity or rating of small underfeed stokers is usually stated as the burning rate in pounds of coal per hour. Codes for establishing uniform methods of rating anthracite and bituminous coal stokers have been adopted by the *Stoker Manufacturers Association*[3].

The Association also has adopted a uniform method of selecting stokers that is published in convenient tables and charts[2]. The required capacity of the stoker is calculated as follows:

$$\frac{\text{Load (Btu per hour)}}{\text{Heating value of coal (Btu per pound)} \times \text{over-all efficiency of stoker and boiler or furnace}} = \text{Stoker burning rate required (pounds of coal per hour)}$$

In determining the total load placed on a stoker-fired boiler by a steam or hot water heating system, a piping and pick-up factor of 1.33 is commonly used in sizing the stoker, but this factor should be increased at times due to unusual conditions.

Controls

The heat delivery from the stoker of the smallest household type to the largest industrial unit can be regulated accurately with fully automatic controls. The smaller heating applications are controlled normally by a

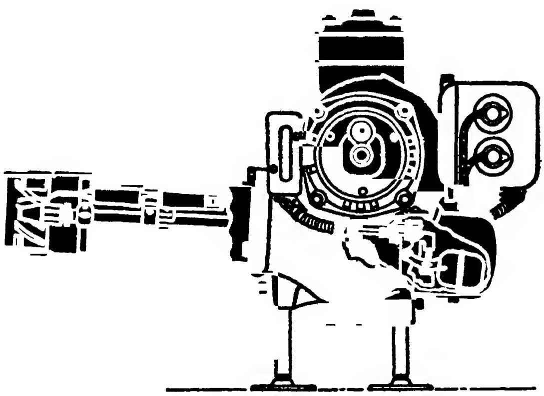

Fig. 12. Low Pressure Atomizing Oil Burner

thermostat placed in the building to be heated. Limit controls are supplied to prevent excessive temperature or pressure being developed in the furnace or boiler and refueling controls are used to maintain ignition during periods of low heat demand. Automatic low water cut-outs are recommended for use with all automatically-fired steam boilers. (See Chapter 34.)

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically from liquid fuels. Two methods are employed for the preparation of the oil for the combustion process; atomization, and vaporization. The simpler types of burners depend upon the natural chimney draft for supplying the air for combustion. Other burners provide mechanical air supply or a combination of atmospheric, and mechanical. Ignition is accomplished by an electrical spark or hot wire, or by an oil or gas pilot. Some burners utilize a combination of these methods. Continuously operating burners may use manual ignition. Burners of different types

operate with luminous or non-luminous flame. Operation may be intermittent, continuous with high-low flame, or continuous with graduated flame.

CLASSIFICATION OF BURNERS

Domestic oil burners may be classified by type of design or operation into the following groups: pressure atomizing or gun, rotary, and vaporizing or pot. These are further classified as mechanical draft, and natural draft.

Pressure Atomizing (Gun Type)

Gun type burners may be divided into two classes, low-pressure, and high-pressure atomization. In the first group, a mixture of oil and primary air is pumped as a spray through the nozzle at a pressure of 2 to 7 lb per square inch. Secondary air is supplied by a fan. Ignition is obtained by means of a high-voltage electric spark used alone, or as primary ignition for a gas pilot. Various features of a low pressure atomizing burner are shown in Fig. 12.

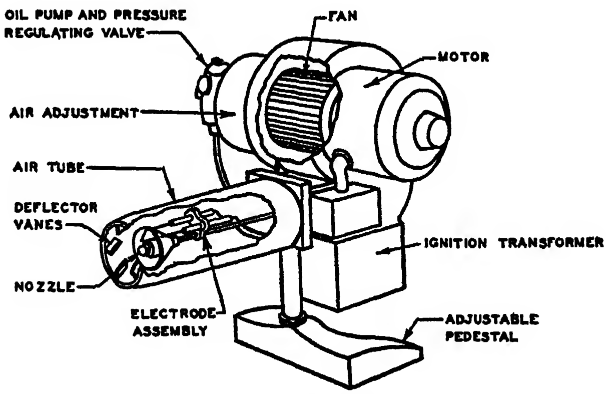

FIG. 13. HIGH-PRESSURE ATOMIZING OIL BURNER

The high-pressure atomizing type, illustrated in Fig. 13, is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and arranged so that a spray of atomized oil is introduced, at about 100 lb per square inch, and mixed in the combustion chamber with the air stream emerging from the air tube. A variety of patented shapes is employed at the end of the air tube to influence the direction and speed of the air and thus the effectiveness of the mixing process.

This type of burner utilizes a fan to supply the air for combustion, and ignition is established by a high-voltage electric spark that may be operative continuously while the burner is running, or just at the beginning of the running period. Gun type burners operate on the intermittent *on-off* principle, and with a luminous flame.

The combustion process is completed in a chamber constructed of refractory material, or stainless steel, this being a part of the installation. Pressure-atomizing burners generally use the distillate oils, No. 1, 2 or 3 grade. (See Chapter 16.)

Rotary Type

This class of burners may be divided into two groups: vertical, and horizontal. Most of the smaller rotary burners are of the vertical type, and use the lighter distillate oils, No. 1 or 2 grade.

The most distinguishing feature of vertical rotary burners is the principle of flame application. These burners are of two general types: the center flame and wall flame. In the former type (Fig. 14), the oil is atomized by being thrown from the rim of a revolving disc or cup and the flame burns in suspension with a characteristic yellow color. Combustion is supported by means of a bowl-shaped chamber or hearth. The wall flame burner (Fig. 15) differs in that combustion takes place in a ring of stainless steel or refractory material, which is placed around the hearth. Dependent upon combustion adjustment, these burners may operate with either a semi-luminous or non-luminous flame.

Both types of vertical rotary burners are further characterized by their installation within the ash pit of the boiler or furnace. Various types of

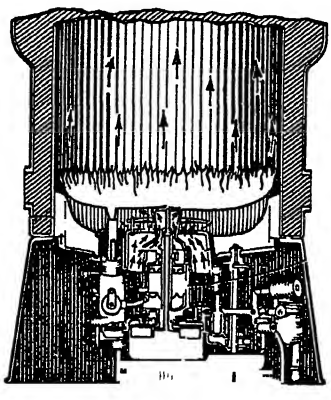

FIG. 14. CENTER FLAME VERTICAL ROTARY BURNER

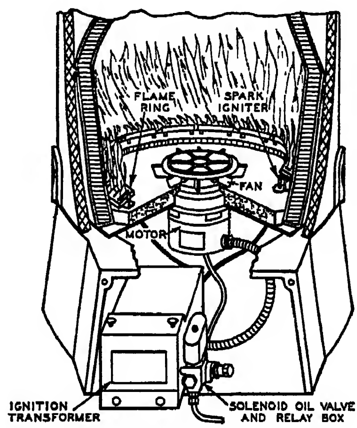

FIG. 15. WALL FLAME VERTICAL ROTARY BURNER

ignition are utilized, gas and electric, either spark or hot wire. The air for combustion is supplied partially by natural draft, and partially by fan effect of the central spinner element.

Horizontal rotary burners are used principally to burn the heavier oils, Nos. 5 and 6 grades, principally in larger commercial and industrial installations, although domestic sizes are available. Such burners are of the mechanical atomizing type, using rotating cups which throw the oil from the edge of the cup at high velocity into the surrounding stream of air delivered by the blower (Fig. 16).

Horizontal rotary burners commonly use a combination electric-gas ignition system, or are lighted manually. Primary air for combustion is supplied by a blower, and secondary air, often introduced through a checkerwork in the combustion chamber, is controlled by chimney draft. These burners operate with a luminous flame, usually on high-low or continuous setting.

In larger installations, burners may be installed in multiple in a common combustion chamber. Because of the high viscosity oils used in these

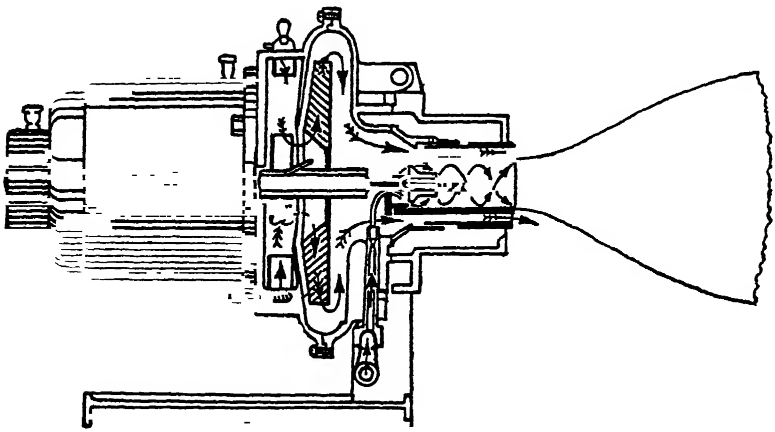

FIG. 16. HORIZONTAL ROTATING CUP OIL BURNER

burners, it is customary to preheat the oil between the tank and the burner. Preheating when delivering from tank car, or truck, is often required in cold weather.

Vaporizing Burners

This type burner transforms the oil into a combustible vapor by the application of heat from a plate, or cracking chamber. The class may be subdivided into burners supplying combustion air mechanically, and those utilizing the natural chimney draft. Vaporizing pot-type burners are generally restricted to the use of the lighter distillates but some wil operate with No. 3 oil.

These burners are designed for continuous or intermittent operation with manual ignition, and the rate of burning is controlled by a metering valve. They are also available with electric ignition and thermostatic controls. Flame may be luminous or non-luminous, dependent upon adjustment. A burner of this type is illustrated in Fig. 17.

Vaporizing burners of the natural draft type are used as the firing device in integral space heaters, water heaters, or furnace units. Some types have also been applied successfully to conversion installations.

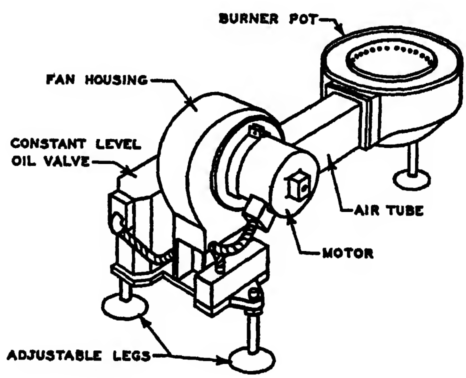

FIG. 17. VAPORIZING POT-TYPE BURNER

Oil-Fired Boiler and Furnace Units

A number of types of specially designed oil-fired boiler-burner and furnace-burner units are available. Various locations of burners will be noted in such units; some having the combustion chamber and burner at the top, some at the bottom, and some at the center of the appliance. One type of boiler-burner unit is shown in Fig. 18. The coordinated design of boiler (or furnace) and burner elements insures the optimum in operating characteristics, and the maintenance of balanced performance. This type of equipment usually has more heating surface, better flue proportions and gas travel than conventional boilers or furnaces. Some

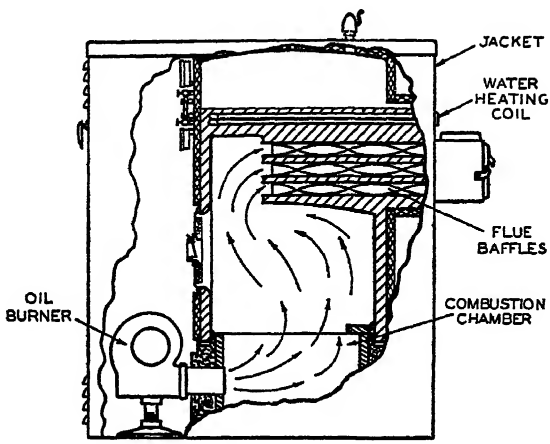

FIG. 18. TYPICAL BOILER-BURNER UNIT

of the better conversion installations, however, may equal the unit type in performance.

Operating Requirements for Mechanical Draft Oil Burners

The *U. S. Department of Commerce* in conjunction with the oil burner industry has established commercial standards for automatic mechanical draft oil burners for domestic installations which cover installation requirements and performance tests[4].

Combustion Process

Efficient combustion must produce a clean flame and use a relatively small excess of air, *i.e.*, between 25 and 50 per cent. This can be done only by vaporizing the oil quickly and completely, and mixing it vigorously with air in a combustion chamber hot enough to support the combustion. A vaporizing burner prepares the oil, for combustion, by transforming the liquid fuel to the gaseous state by the application of heat. This is accomplished before the oil vapor mixes with air to any extent and, if the air and oil vapor temperatures are high and the fire pot hot, a clear blue flame is produced.

In an atomizing burner the oil is mechanically separated into very fine particles so that the surface exposure of the liquid to the radiant heat of the combustion chamber is vastly increased and vaporization proceeds quickly. The result is the ability to burn more and heavier oil within a given combustion space. Because the air enters the combustion chamber

with the liquid fuel particles, mixing, vaporization and burning occur all at once in the same space. This produces a luminous flame. A deficient amount of air is indicated by a dull red or dark orange flame with smoky tips.

An excessive supply of air may produce a brilliant white flame or a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be detected, it is not possible to distinguish, by eye, the effect of the finer adjustment which competent installation requires.

Combustion Adjustments

The present-day oil burner with mechanical oil and air supply, properly installed and equipped with an automatic draft regulator, is capable of maintaining efficient combustion for a considerable period following the initial adjustments of oil and air. Eventually certain changes may occur, however, that will cause the per cent of excess air to decrease below allowable limits. A decrease in air supply while the oil delivery remains constant, or an increase in oil delivery while the air supply remains constant, will make the mixture of oil and air too rich for clean combustion. The more efficient the adjustment the more critical it will be. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (1) changes in oil viscosity due to temperature change or variations in grade of oil delivered, (2) erosion of atomizing nozzle, (3) fluctuations in by-pass relief pressures, and (4) possible variations in methods of atomization. Any change due to partial stoppage of oil delivery will increase the proportion of excess air. This will result in less heat, reduced economy and possibly a complete interruption of service.

The following factors may influence the air supply: (1) changes in combustion draft due to a variety of causes (*i.e.*, changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney, and changes in draft resistance of boiler due to partial stoppage of the flues), and (2) changes in air inlet adjustments at the fan.

Air leakage into the boiler or furnace setting should be reduced to a minimum. The amount of air leakage will be determined by the draft in the combustion chamber. It is important that this draft should be reduced as low as is consistent with the proper disposal of the gases of combustion. When using mechanical draft burners with average conditions, the combustion chamber draft should not be allowed to exceed 0.02-0.05 in. water. An automatic draft regulator is very helpful in maintaining such values.

Even though a fan is generally used to supply the air for combustion, in most oil burners, the importance of a proper chimney should not be overlooked. The chimney should have sufficient height and size to insure that the draft will be uniform within the limits given if maximum efficiency throughout the heating season is to be maintained.

Measurement of the Efficiency of Combustion

Since efficient combustion is based upon a clean flame and definite proportions of oil and air employed, it is possible to determine the results by analyzing the combustion gases. It is usually sufficient to analyze only for carbon dioxide (CO_2). A showing of 10 to 12 per cent indicates

the best adjustment if the flame is clean. Most of the good installations show from 8 to 10 per cent CO_2. Taking into account the potential hazard of low excess air (high CO_2), a setting to give 10 per cent CO_2 constitutes a reasonable standard for most oil burners.

Combustion Chamber Design

With burners requiring a refractory combustion chamber the size and shape should be in accordance with the manufacturer's instructions. It is important that the chamber shall be as nearly air tight as is possible, except when the particular burner requires a secondary supply of air for combustion.

The atomizing burner is dependent upon the surrounding heated refractory or firebrick surfaces to vaporize the oil and support combustion. Unsatisfactory combustion may be due to inadequate atomization and mixing. A combustion chamber can only compensate for these things to a limited extent. If liquid fuel continually reaches some part of the firebrick surface, a carbon deposit will result. The combustion chamber should enclose a space having a shape similar to the flame but large enough to avoid flame contact. The nearest approach in practice is to have the bottom of the combustion chamber flat, but far enough below the nozzle to avoid flame contact, the sides tapering from the air tube at the same angle as the nozzle spray and the back wall rounded. A plan view of the combustion chamber resembles in shape the outline of the flame. In this way as much firebrick as possible is close to the flame so it may be kept hot. This insures quick vaporization, rapid combustion and better mixing by eliminating dead spaces in the combustion chamber. An overhanging arch at the back of the fire pot is sometimes used to increase the flame travel and give more time for mixing and burning, and sometimes to prevent the gases from going too directly into the boiler flues. When good atomization and vigorous mixing are achieved by the burner, combustion chamber design becomes a less critical matter. Where secondary air is used, combustion chamber design is quite important. When installing some of the vertical rotary burners the manufacturer's instructions must be followed carefully when installing the hearth, as in this class successful performance depends upon this factor.

Boiler Settings

As the volume of space available for combustion is a determining factor in oil consumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the ash pit volume; in new installations the boiler may be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2.5 lb of oil per hour can properly be burned. This corresponds to an average liberation of about 38,000 Btu per cubic foot per hour. At times much higher fuel rates may be satisfactory. For best results, care should be taken to keep the gas velocity below 40 fps. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or firebrick surfaces. Manufacturers of

oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

Controls

Controls for oil burner operation, including devices for the safety and protection of a boiler or furnace, are fully described in Chapter 34.

GAS-FIRED HEATING EQUIPMENT

A gas burner is defined by the *American Gas Association* as "a device for the final conveyance of the gas, or a mixture of gas and air, to the combustion zone." Burners used for domestic heating are of the atmospheric injection, yellow flame, or power burner types.

The use of gas has resulted in the production of a number of types of domestic gas heating appliances, and systems. These may be classified

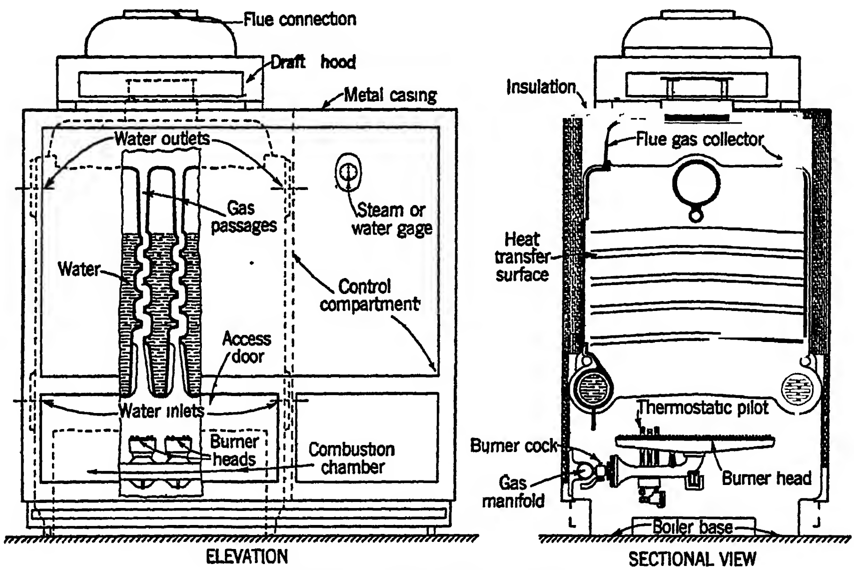

Fig. 19. Gas-Fired Boiler

in types designed for central heating plants, and those for unit application. Gas-designed units and conversion burners are available for the several kinds of central systems. Unit heaters, space heaters and circulators may be had for installation in the space being heated.

Central Heating Systems

Boilers and furnaces specially designed for gas-firing incorporate design features for obtaining maximum efficiency and performance. Small flue passes to secure good heat transfer, the use of materials resistant to the corrosive effects of products of combustion, and draft hoods are notable features. Control equipment includes gas pressure regulators, thermostatic pilots, and limit controls designed to protect the appliance and to insure safety of operation. A boiler designed for gas-burning is illustrated in Fig. 19.

Conversion burners are usually complete burner and control units designed for installation in existing boilers and furnaces. Burner heads

are of circular or rectangular shape in order to fit in the space available. The control equipment is generally the same as for gas boilers and furnaces. Various baffles made of clay radiants or metal are used for the purpose of guiding the products of combustion along the heating surface in the firebox or flues. Automatic air dampers are supplied on many models to prevent flow of air into the firebox when the burner is not operating. A typical gas conversion burner is shown in Fig. 20.

Burners of this type are available in sizes ranging from 80,000 to 500,000 Btu per hour capacity. Burners of even larger capacity, for use with natural gas in large steel boilers, are usually engineered by the local utility or contractor. They are available in an infinite number of sizes because the burner may be an assembly of multiple burner heads filling the entire firebox.

Domestic sizes of conversion burners should be installed with due attention to the method of venting. Draft hoods, conforming to American Standard Requirements, should be installed in place of the dampers used with a solid fuel.

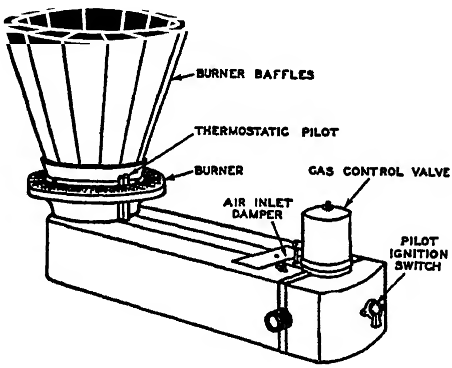

FIG. 20. TYPICAL GAS CONVERSION BURNER

One form of central heating system is the warm air floor furnace [5]. The use of these furnaces is adaptable to mild climates. They are used for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility. With the usual type the register is installed in the floor, the heating element and gas piping being suspended below.

Unit Type Heaters

Space heaters may be used for auxiliary heating, but in many cases are installed for furnishing heat to entire buildings. With the exception of wall heaters, they are semi-portable.

Parlor heaters or *circulators* are usually of the cabinet type. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base, passes up inside the jacket around a heating section, and out of the heater at, or near, the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation.

The burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass around baffles within the heating element, and out the flue at the back near the top. They are well adapted for residence room heating and also for stores and offices.

Radiant heaters give off a considerable portion of their heat in the form of radiant energy emitted by an incandescent refractory that is heated by a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to fourteen or more radiants. An atmospheric burner is supported near the center of the base. Others have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are portable; however, there are also types which are encased in a jacket that fits into the wall with a grilled front.

Gas-fired steam and hot water radiators are other types of room heating appliances. They are made in a large variety of shapes and sizes and are similar in appearance to the ordinary steam or hot water radiator. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to steam or hot water radiators. They are usually constructed of sheet metal hollow sections. The products of combustion circulate through the sections and are discharged from a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Unit heaters are used extensively for heating large spaces such as stores, garages, and factories. These heaters consist of a burner, heat exchanger, fan for distributing the air, draft hood, thermostatic pilot, and controls for burners and fan. They are usually mounted in an elevated position from which the heated air is directed downward by louvers. Some unit heaters are suspended from the ceiling, and others are free-standing floor units of the heat tower type.

Unit heaters are available in two types, classified according to their use, with, or without ducts. Only those types of unit heaters tested and approved as warm air furnaces can be connected safely to ducts, as they have sufficient blower capacity to deliver an adequate air supply against duct resistance and are equipped with limit controls.

Combustion Process and Adjustments

Most domestic gas burners are of the atmospheric injection (Bunsen) type in which primary air is introduced, and mixed with the gas in the throat of the mixing tube. A ratio of about 3 parts primary air to 1 part gas for manufactured gas, and a 5½ to 1 ratio for natural gas, are generally used as theoretical values. The amount of excess air required in practice depends upon several factors, notably; uniformity of air distribution and mixing, direction of gas travel from burner, and the height and temperature of combustion chamber.

Secondary air is drawn into gas appliances by natural draft. As with other fuels, excess secondary air constitutes a loss, and should be reduced to a proper minimum, which usually cannot be less than 25 to 35 per cent if the appliance is to meet *A.S.A.* approval. Yellow flame burners depend upon secondary air, alone, for combustion.

The flame produced by atmospheric injection burners is non-luminous. Air shutter adjustments for manufactured gas should be made by closing

the air shutter until yellow flame tips appear and then by opening the air shutter to a final position at which the yellow tips just disappear. This type of flame obtains ready ignition from port to port and also favors quiet flame extinction. When burning natural gas the air adjustment is generally made to secure as blue a flame as obtainable.

Little difficulty should be had in maintaining efficient combustion when burning gas. The fuel supply is normally held to close limits of variation in pressure and calorific value and the rate of heat supply is nominally constant. Because the force necessary to introduce the fuel into the combustion chamber is an inherent factor of the fuel, no draft by the chimney is required for this purpose. The use of a draft hood insures the maintenance of constant low draft condition in the combustion chamber with a resultant stability of air supply. A draft hood is also helpful in controlling the amount of excess air and preventing back drafts that might extinguish the flame. (See Chapter 16.)

Due to the use of draft hoods and gas pressure regulators both the input and combustion conditions of gas appliances are maintained quite uniform until deposits of dirt, corrosion, or scale accumulate in the air inlet openings, burner ports, or on the heating surface. Periodic cleaning is necessary to keep any gas appliance in proper operating condition.

Measurement of the Efficiency of Combustion

The efficiency of combustion may be judged from the percentage of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. The CO_2 and O_2 may be obtained by means of an Orsat apparatus but the CO must be determined by more accurate equipment. It is customary to use simple indicators to determine whether CO is present and to make adjustments of the appliances to reduce the CO below 4/100 of one per cent before continuing tests in which the CO_2 and O_2 can then be found by use of the Orsat apparatus. Since the ultimate CO_2 for any gas depends on the total hydrogen content the quality of the combustion should not be judged from the value of the CO_2 in the flue gas without reference to the ultimate CO_2 obtainable. Practical values of CO_2 will usually be from 8 to 14 per cent depending on the gas used.

Ratings for Gas Appliances

Input rating for a gas appliance is established by demonstrating that the appliance can meet the Approval Requirements of the *A.S.A.* The tests are conducted at the *A.G.A. Testing Laboratories.* Output rating is determined from the approved input and an average efficiency stated in the Approval Requirements and is the heat available at the outlet.

Sizing Gas-Fired Heating Plants

Although gas-burning equipment usually is completely automatic, maintaining the temperature of rooms at a predetermined figure, there are some manually controlled installations. In order to overcome effectively the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 per cent greater than the equivalent standard radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up loads and consequently, it is possible to use a lower selection factor. For a gas-fired boiler or furnace under thermostatic control a factor of 20 to 25 per cent is usually sufficient for pick-up allowance.

In those installations, in mild climates where 100 per cent outside air is used, furnaces should be of larger size in order to provide adequate capacity and quick pick-up under intermittent heating conditions.

The factor to be allowed for loss of heat from piping will vary somewhat, the proportionate amount of piping installed being greater for small installations than for large ones. For selection factors to be added to installed radiation under thermostatic control see Chapter 18.

Appliances used for heating with gas should bear the approval seal of the *A.G.A. Testing Laboratories* on the manufacturer's nameplate, together with the official input and output ratings. It is not permissible to operate a gas heating unit above its stated rating. It may be necessary to operate below this rating at elevations above 3,000 ft.

Installations should be made in accordance with recommendations shown in the publications of the *American Gas Association.*

Controls

Temperature controls for gas burners are described in Chapter 34. Some central heating plants are equipped with push-button or other manual control. The main gas valve may be of either the snap action or throttling type. Automatic electric ignition is available.

FUEL BURNING RATES

The burning rate for automatic fuel burning devices is determined by the gross heat output required of the boiler, or furnace, to carry the net heating load plus allowances for system losses, and pick-up. General values for these allowances previously have been noted. Detailed information for piping and pick-up allowances for steam, and hot water systems is given in Chapter 18 and for warm air systems in Chapters 21 and 22.

When the gross output, operating efficiency, and heat value of the fuel are known, the required rate of burning can be determined by means of Figs. 21, 22 and 23 for the several fuels.

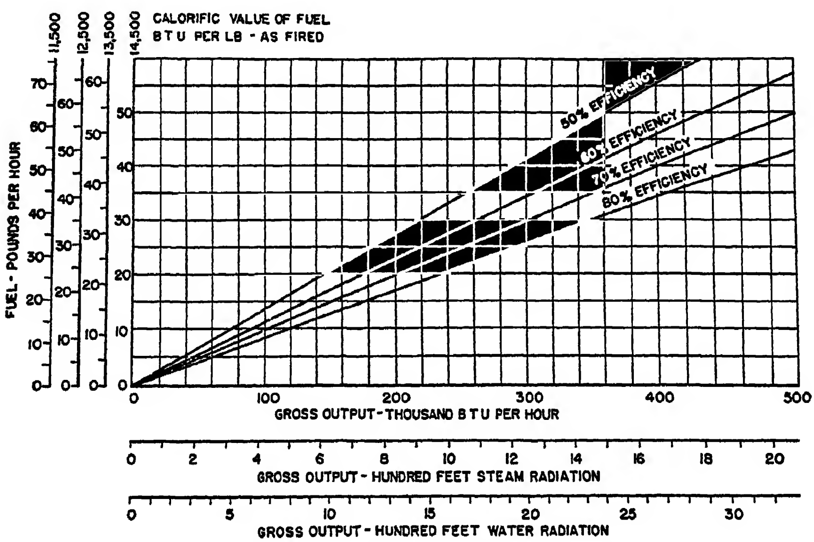

Fig. 21. Coal Fuel Burning Rate Chart

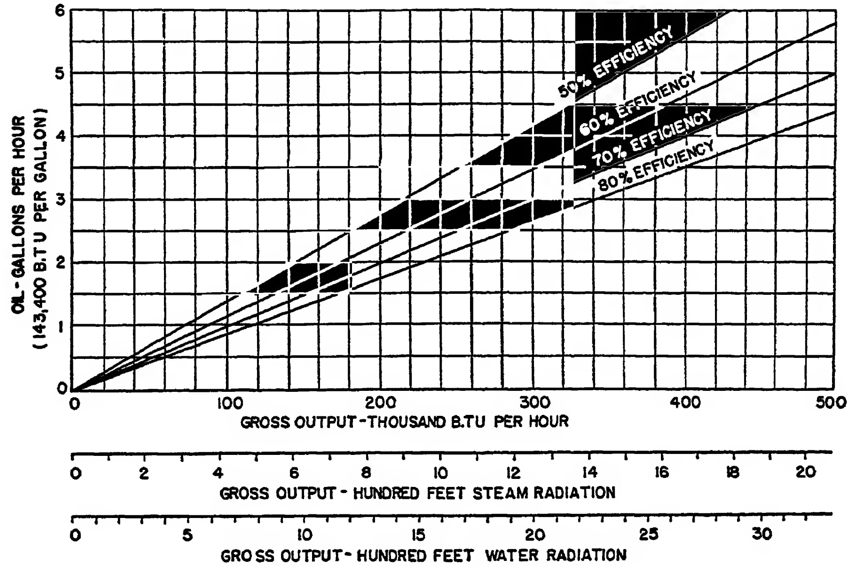

FIG. 22. OIL FUEL BURNING RATE CHART[a]

[a]This chart is based upon No 3 oil having a heat content of 143,400 Btu per gallon. If other grades of oil are used multiply the value obtained from this chart by the following factors: No. 1 oil (139,000 Btu per gallon) 1.032; No. 2 oil (141,000 Btu per gallon) 1 017; No 4 oil (144,500 Btu per gallon) 0.992; No. 5 oil (146,000 Btu per gallon) 0.982; and No. 6 oil (150,000 Btu per gallon) 0.956.

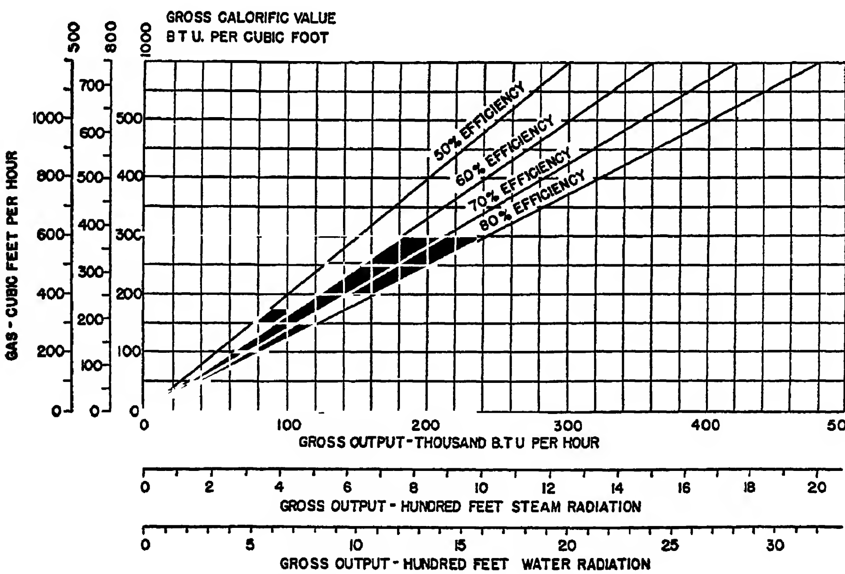

FIG. 23. GAS FUEL BURNING RATE CHART

As the rate of fuel burning is directly proportional to the load for a given efficiency, these charts can be extended by moving the decimal points the same number of digits in both vertical and horizontal scales.

The correct fuel burning rate can be determined directly from the several charts for oil or gas burning installations, as these customarily operate on a strictly intermittent basis. These fuel burning devices usually introduce the fuel at a single fixed rate during the *on* periods and this rate should be sufficient to carry the gross load. In the case of coal stokers, which are usually capable of variable rates of firing, it is desirable to operate at as low a rate as weather conditions will permit, but the maximum firing rate of the stoker should be sufficient to carry the gross load. This rate may be determined by the same method as used for oil or gas.

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[1]—Domestic Burners for Pennsylvania Anthracite (Underfeed Type), (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS48-40).

[2]—*Stoker Manufacturers Association Manual:* Industry Standards, Recommended Practices, Technical Information. Published by *Stoker Manufacturers Association*, 307 N. Michigan Ave., Chicago 1, Ill.

[3]—Code for Determination of Rated Capacities of Anthracite Underfeed Stokers, adopted June 1, 1944, and a Code for Determination of Rated Capacities of Bituminous Underfeed Stokers, adopted May 8, 1944. See *Stoker Manufacturers Association Manual.*

[4]—Automatic Mechanical Draft Oil Burners Designed for Domestic Installations (*U. S Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS75-42). Flue Connected Oil Burning Space Heaters Equipped with Vaporizing Pot Type Burners (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS101-43). Warm-Air Furnaces Equipped with Vaporizing Pot-Type Oil Burners (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS(E)104-43).

[5]—Gas Floor Furnaces, Gravity Circulating Type (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS99-42).

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Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 185).

CHAPTER 18

Heating Boilers and Furnaces

BOILERS: Construction, Types, Design Considerations, Testing and Rating Codes, Efficiency, Rating, Selection, Space Limitations, Connections and Fittings, Erection, Operation and Maintenance. FURNACES: Types, Materials and Construction, Ratings, Testing and Rating Codes, Efficiency, Design Considerations, Humidification Equipment

IN presenting the subject of Boilers and Furnaces this chapter is divided into two parts; the first dealing with boilers and the second treating warm air furnaces.

HEATING BOILERS

Steam and hot water boilers for low pressure heating are built in a wide variety of types and sizes, many of which are illustrated in the *Catalog Data Section.* They are made of steel or cast-iron.

CONSTRUCTION

The only code governing the construction of low-pressure heating steel and cast-iron boilers that has gained recognition on a national basis is the *ASME* Boiler Construction Code for Low Pressure Heating Boilers. Some states and municipalities have their own codes which apply locally but these are usually patterned after the *ASME* Code.

The maximum allowable working pressures are limited by the *ASME* Code to 15 psi for steam and 30 psi for hot water heating boilers. Hot water boilers may be used for higher working pressures, for heating purposes, or for hot water supply when designed and tested for the higher pressure.

TYPES OF HEATING BOILERS

Heating boilers are classified in a number of different ways, such as:

(*a*) According to materials of construction. These are steel and cast-iron. Very few non-ferrous boilers are made.

(*b*) According to the fuels for which the boilers are designed. These are coal, hand fired or stoker fired; oil; gas; or wood. Some boilers are designed specifically for one fuel but many boilers are designed for more than one fuel.

(*c*) According to the specific purpose or application for which the boiler is used, such as space heating or domestic hot water supply.

(*d*) According to the design or construction of the boiler such as, sectional, round, fire-tube, water-tube, magazine feed, Scotch, etc.

Cast-Iron Boilers

Cast-iron boilers are generally classified as:

(*a*) Square or rectangular boilers with vertical sections and rectangular grates commonly known as sectional boilers.

(*b*) Round boilers with horizontal pancake sections and circular grates.

Cast-iron boilers are usually shipped in sections and assembled at the place of installation. In the majority of boilers the sections are assembled with push nipples and tie rods. Many sectional boilers are provided with

large push nipples at top to permit the circulation of water between adjacent sections at both the water line and bottom of the boiler, which is necessary to enable the use of an indirect water heater with the boiler for summer-winter hot water supply. Round and sectional boilers may be increased in size by the addition of sections and corresponding plate work.

Small sectional type boilers are available with *wet-base* construction, wherein the ashpit or combustion chamber sides and bottom are surrounded by extensions of the water legs of the boiler sections and thus no separate base is required. This type of construction permits the boiler to be set directly on a wood or composition floor without danger of fire. The *wet-base* also provides some additional heating surface.

Capacities of cast-iron boilers range generally from capacities required for small residences up to about 12,000 sq ft of steam radiation. There are a few boilers made with capacities up to 18,000 sq ft of steam radiation. For larger loads, boilers must be installed in multiple.

Steel Boilers

Steel boilers may be of the fire-tube type, in which the gases of combustion pass through the tubes and the boiler water circulates around them, or of the water-tube type, in which the gases circulate around the tubes and the water passes through them.

Either the fire-tube or water-tube type may be designed with integral water jacketed furnaces or arranged for refractory lined brick or refractory lined jacketed furnaces. Those with integral water jacketed furnaces are called portable firebox boilers and are the most commonly used type. They are usually shipped in one piece, ready for piping. Refractory furnaces are usually installed in refractory lined furnace boilers after they are set in place.

Capacities of steel boilers range from those required for small residences up to about 35,000 sq ft of steam radiation.

Boilers for Special Applications

One of these is known as the *magazine feed boiler* developed for the burning of small sizes of anthracite and coke and has a large fuel carrying capacity which results in longer firing periods than would be the case with the standard types burning coal of buckwheat size. Special attention must be given to proper chimney sizes and connections when installing in order to insure adequate draft.

Boilers for *hot water supply* are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-forming properties of the water supplied and the temperatures maintained. If low water temperatures are maintained the life of the heater will be much longer due to decreased scale formation and minimized corrosion. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.

2. The boiler is protected from scale and corrosion.

3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency although it will affect the capacity of the heat exchanger.

4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam or a forced circulation hot water heating system is installed, the domestic hot water may be heated by an indirect heater attached to the boiler. For most satisfactory performance in the steam system, this heater is placed just below the water line of the boiler. In a forced circulation hot water system, it should be located as high as possible with respect to the boiler.

BOILER DESIGN CONSIDERATIONS

Furnace Design

Good efficiency and proper boiler performance are dependent on correct furnace design. There must be sufficient volume for burning the particular fuel which is used, and means to obtain a thorough mixing of air and gases at a high temperature and at a velocity low enough to permit complete combustion of all the volatiles. For hand fired boilers, the furnace volume should be large enough to hold sufficient fuel for reasonably long firing periods. (See Chapters 16 and 17.)

Heating Surface

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines is known as *direct* or radiant surface and that in contact with hot gases only as *indirect* or convection surface. The amount of heating surface, its distribution, and the temperatures on either side thereof influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. The area of the gas passages must not be so small as to cause excessive resistance to the flow of gases where natural draft is employed. Inserting baffles so that the heating surface is arranged in series with respect to the gas flow increases boiler efficiency and reduces stack temperature, but increases the draft loss through the boiler.

Heat Transfer Rate

Practical average over-all heat transfer rates expressed in Btu absorbed per square foot of surface per hour will average about 3300 for hand fired boilers and 4000 for mechanically fired boilers when operating at *design load*. When mechanically fired boilers are operating at *maximum load* as defined in this chapter under heading *Selection of Boilers*, these values will run between 5000 and 6000. Boilers operating under favorable conditions at these heat transfer rates will give exit gas temperatures that

are considered consistent with good practice, although there are boilers which have high efficiencies and also operate at higher transmission rates.

TESTING AND RATING CODES

The Society has adopted four solid fuel testing codes, a solid fuel rating code, and an oil fuel testing code.

ASHVE Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June, 1929) [1], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics.

ASHVE Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)[1] is intended for use with ASHVE Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers [2]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler.

The ASHVE Standard Code for Testing Steam Heating Boilers Burning Oil Fuel [3], (Adopted June, 1932), is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers.

In 1938 the Society adopted a Standard Code for Testing Stoker-Fired Steam Heating Boilers [4], (Adopted June, 1938), which is intended to provide a test method for determining the efficiency and performance characteristics of any stoker and boiler combination burning any type of solid fuel such as anthracite or bituminous coal.

The *Steel Boiler Institute, Inc.* has adopted (June 12, 1945) a Rating Code for Commercial Steel Boilers and Residential Steel Boilers and for Testing Oil-Fired Residential Steel Boilers. The commercial boilers (defined as those having 129 to 2500 sq ft of heating surface) are rated in

Table 1. *SBI* Net Rating Data for Residential Steel Boilers—Oil Fired[a]

SBI Net Rating			Minimum Furnace Volume Cu Ft	Heating Surface Sq Ft
Sq Ft Steam	Sq Ft Water	Btu		
275	440	66000	2.5	16
320	510	77000	2.9	19
400	640	96000	3.6	24
550	880	132000	5.0	32
700	1120	168000	6.4	41
900	1440	216000	8.2	53
1100	1760	264000	10.0	65
1300	2080	312000	11.8	77
1500	2400	360000	13.6	88
1800	2880	432000	16.4	106
2200	3520	528000	20.0	129
2600	4160	624000	23.6	153
3000	4800	720000	27.3	177

[a]Stoker-fired and Gas-fired *SBI* Net Rating not greater than Oil-fired. Hand-fired, *SBI* Net Rating (Steam) not greater than 14 times the square feet of heating surface.

Table 2. *SBI* Ratings for Commercial Steel Boilers

Mechanically Fired								Heating Surface Sq Ft	Hand Fired						Minimum Grate Area Sq Ft	Flow Tappings	
SBI Rating			*SBI* Net Rating			Minimum Furnace Volume Cu Ft	Minimum Furnace Height[a] In.		*SBI* Rating			*SBI* Net Rating				Steam Outlet I.P.S. In.	Steam Return I.P.S. In.
Sq Ft Steam	Sq Ft Water	Btu	Sq Ft Steam	Sq Ft Water	Btu				Sq Ft Steam	Sq Ft Water	Btu	Sq Ft Steam	Sq Ft Water	Btu			
2,190	3,500	526,000	1,800	2,880	432,000	15.7	26	129	1,800	2,880	432,000	1,500	2,400	360,000	7.9	6	3
2,680	4,280	643,000	2,200	3,520	528,000	19.2	28	158	2,200	3,520	528,000	1,830	2,930	439,000	8.9	6	3
3,160	5,050	758,000	2,600	4,160	624,000	22.6	29¼	186	2,600	4,160	624,000	2,170	3,470	521,000	9.7	6	3
3,650	5,840	876,000	3,000	4,800	720,000	26.1	29½	215	3,000	4,800	720,000	2,500	4,000	600,000	10.5	6	3
4,250	6,800	1,020,000	3,500	5,600	840,000	30.4	30	250	3,500	5,600	840,000	2,920	4,670	700,000	11.4	6	3
4,860	7,770	1,166,000	4,000	6,400	960,000	34.8	30½	286	4,000	6,400	960,000	3,330	5,330	800,000	12.2	6	3
5,470	8,750	1,313,000	4,500	7,200	1,080,000	39.1	31¼	322	4,500	7,200	1,080,000	3,750	6,000	900,000	13.4	6	3
6,080	9,720	1,459,000	5,000	8,000	1,200,000	43.5	31¾	358	5,000	8,000	1,200,000	4,170	6,670	1,000,000	14.5	6	3
7,290	11,660	1,750,000	6,000	9,600	1,440,000	52.1	32¾	429	6,000	9,600	1,440,000	5,000	8,000	1,200,000	16.4	8	4
8,500	13,600	2,040,000	7,000	11,200	1,680,000	60.8	34	500	7,000	11,200	1,680,000	5,830	9,330	1,400,000	18.1	8	4
10,330	16,520	2,479,000	8,500	13,600	2,040,000	73.8	35½	608	8,500	13,600	2,040,000	7,080	11,330	1,700,000	20.5	8	4
12,150	19,440	2,916,000	10,000	16,000	2,400,000	86.8	37½	715	10,000	16,000	2,400,000	8,330	13,330	2,000,000	22.5	8	4
15,180	24,280	3,643,000	12,500	20,000	3,000,000	108.5	40¼	893	12,500	20,000	3,000,000	10,420	16,700	2,500,000	25.6	8	4
18,220	29,150	4,373,000	15,000	24,000	3,600,000	130.2	43	1,072	15,000	24,000	3,600,000	12,500	20,000	3,000,000	28.4	8	4
21,250	34,000	5,100,000	17,500	28,000	4,200,000	151.8	46	1,250	17,500	28,000	4,200,000	14,580	23,330	3,500,000	30.9	8	4
24,290	38,860	5,830,000	20,000	32,000	4,800,000	173.5	48¾	1,429	20,000	32,000	4,800,000	16,670	26,670	4,000,000	33.2	8	4
30,360	48,570	7,286,000	25,000	40,000	6,000,000	216.9	54½	1,786	25,000	40,000	6,000,000	20,830	33,330	5,000,000	37.4	10	6
36,430	58,280	8,743,000	30,000	48,000	7,200,000	260.3	60½	2,143	30,000	48,000	7,200,000	25,000	40,000	6,000,000	41.2	10	6
42,500	68,000	10,200,000	35,000	56,000	8,400,000	303.6	66	2,500	35,000	56,000	8,400,000	29,170	46,670	7,000,000	44.7	10	6

[a]Bituminous Stoker Fired.

square feet (steam) on the basis of heating surface with limitations set for grate area, furnace volume, and furnace height. The residential boilers (defined as those having not more than 177 sq ft of heating surface) are rated on tests for oil-fired boilers, with limitations in relation to heating surface and testing conditions. Stoker-fired and gas-fired *residential* boilers are rated (*SBI* Net Rating) not in excess of the oil-fired rating. Hand-fired residential boilers are rated (*SBI* Net Rating) not greater than 14 times the heating surface.

Tables 1 and 2 show the *SBI* ratings of residential and commercial steel boilers respectively.

The *Institute of Boiler and Radiator Manufacturers* has adopted a Code [5] for rating cast-iron heating boilers based upon performance obtained under controlled test conditions. This Code applies to all sectional cast-iron heating boilers except those of magazine feed type.

The *Gross I=B=R* Output is obtained by test and is subject to certain limiting factors. For hand fired boilers, the number of boilers of a series to be tested, the minimum over-all efficiency, the minimum time limit (the time an *Available Fuel Charge* will last when burned at a rate which will produce the *Gross I=B=R Output*), the chimney area and height, and the draft in the stack are all subject to the limits established in the Code. Tests are run using anthracite coal of standard specification. Bituminous coal and coke ratings are the same as for anthracite coal.

For automatically fired boilers, the number of boilers of a series to be tested, the flue gas temperature and analysis, the minimum over-all efficiency, the draft loss through the boiler, and the heat release in the combustion chamber are subject to limitation by the Code [5]. Automatically fired boiler ratings are established by oil fired tests using gun type oil burners and commercial grade No. 2 fuel oil. Stoker fired and gas fired ratings (where no *A.G.A*. Rating is published) are based on the *Gross I=B=R Output* obtained by oil fired tests.

The *Net I=B=R Rating* is determined from the *Gross I=B=R Output* by applying specified *Piping and Pickup Factors* which range from 2.36 to 1.40 for hand fired boilers and from 1.56 to 1.288 for automatically fired boilers. In both cases, the factor decreases as the boiler size increases. Table 3 is abstracted from the *I=B=R Rating Tables* in the Code and illustrates the relationship between *Net I=B=R Rating* and *Gross I=B=R Output.*

The *American Gas Association* has adopted a method of rating gas designed boilers based upon performance under tests. This is described in Approval Requirements for Central Heating Gas Appliances.

The *Heating, Piping and Air Conditioning Contractors National Association* has adopted a method of rating boilers based on their physical characteristics for those boilers that are not rated in accordance with the *SBI* or *I=B=R* Codes. Ratings are expressed on a Net Load basis in square feet of steam radiation.

BOILER EFFICIENCY

The term *efficiency* as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace*

TABLE 3. $I=B=R$ RATING TABLE

Net $I=B=R$ Rating		Piping Factor	Hand-Fired					Automatic-Fired			
Sq Ft Steam	1000 Btu		Piping and Pickup[a] Factor	Gross $I=B=R$ Output 1000 Btu	Time Available Fuel Will Last, Hr	Maximum Stack Height Ft	Minimum Stack Area Sq In.	Piping and Pickup[a] Factor	Gross $I=B=R$ Output 1000 Btu	Minimum Stack Area Sq In.	Maximum Allowable Draft Loss
1	2	3	4	5	6	7	8	9	10	11	12
100	24 0	1 300	2.360	56 6	7.50	29 0	50.0	1 560	37.4	50 0	0.044
250	60 0	1 293	2.341	140 5	6 77	33 0	50.0	1.548	92.9	50 0	0.051
400	96 0	1 275	2.268	217 7	6 32	36 5	50.0	1 525	146 4	50 0	0.058
550	132 0	1 260	2.201	290 5	5 99	39 5	50.0	1 507	198.9	50 0	0 065
700	168 0	1 248	2.139	359 4	5 73	42 0	54 0	1 492	250.7	50 0	0.072
850	204 0	1 236	2 084	425 1	5 50	44 0	68.5	1.478	301.5	50 0	0.078
1000	240 0	1 225	2 039	489 4	5 32	46 0	82.0	1.466	351 8	52 0	0.084
1150	276	1 215	2 001	552	5 15	47.5	95.5	1.454	401	63 0	0.090
1300	312	1 205	1 967	614	4 99	49 0	110.0	1.444	451	73.5	0 096
1450	348	1 198	1 939	675	4.85	50.5	122.5	1.434	499	84.0	0 102
1600	384	1 190	1.914	735	4.73	52.0	135 0	1 424	547	95 0	0 108
1750	420	1.182	1.891	794	4 60	53 0	147 0	1 416	595	105 0	0.113
1900	456	1 177	1.872	854	4 49	54.5	160 0	1 408	642	115 0	0 118
2050	492	1 169	1.855	913	4 41	55.5	172 0	1 401	689	125 0	0 124
2200	528	1.162	1.837	970	4 34	56 5	185 0	1.394	736	135 0	0 130
2350	564	1.158	1 821	1027	4.29	58 0	196.0	1 388	783	145 0	0.136
2500	600	1 152	1.805	1083	4 24	59 0	207.5	1.382	829	155 0	0 141
2650	636	1 148	1.789	1138	4.19	60.0	219.0	1.376	875	165 0	0.146
2800	672	1 142	1.774	1192	4.15	61.0	231.0	1.369	920	173.5	0 152
2950	708	1 139	1.759	1245	4.11	62.0	242.0	1.364	966	183.5	0.157
3100	744	1 136	1 744	1298	4.07	63.0	253 0	1.359	1011	192.5	0.162
3250	780	1.132	1.730	1349	4 03	64.0	264.0	1.354	1056	202.5	0.167
3400	816	1 129	1.717	1401	4 00	64.5	275 0	1.349	1101	211.5	0 172
3550	852	1.128	1.704	1452	4.00	65.5	286 0	1.344	1145	221.0	0.178
3700	888	1.122	1 691	1502	4 00	66 5	296 0	1.339	1189	230.0	0.183
3850	924	1.121	1.678	1550	4 00	67.5	306 0	1.335	1234	239.5	0.188
4000	960	1.120	1.666	1599	4 00	68 0	315 0	1.331	1278	249.0	0 192
4200	1008	1.120	1.650	1663	4 00	69 0	326 0	1 325	1336	261.5	0.200
4400	1056	1.120	1.634	1726	4.00	70 0	337.0	1 320	1394	274.0	0.206
4600	1104	1.120	1.620	1788	4 00	70.5	349 0	1 314	1451	285.5	—
4800	1152	1 120	1.605	1849	4.00	71.5	358 0	1.310	1509	297 0	—
5000	1200	1 120	1 590	1908	4 00	72 5	367.0	1 305	1566	308 0	—
5200	1248	1 120	1 577	1968	4 00	73 0	376.0	1 301	1624	318 5	—
5400	1296	1.120	1.564	2027	4 00	74.0	385 0	1 297	1681	329 5	—
5600	1344	1.120	1.552	2086	4 00	74 5	393.0	1 294	1739	339 0	—
5800	1392	1.120	1 539	2142	4 00	75.5	402.0	1 291	1797	350 0	—
6000	1440	1 120	1.526	2197	4.00	76 0	409 0	1 290	1858	359 0	—
6200	1488	1.120	1.514	2253	4.00	77.0	417	1.289	1918	369	—
6400	1536	1.120	1.502	2307	4.00	77.5	425	1 288	1978	377	—
6600	1584	1.120	1 491	2362	4.00	78 0	431	1 288	2040	386	—
6800	1632	1 120	1 480	2415	4.00	79 0	438	1 288	2102	395	—
7000	1680	1.120	1 470	2470	4 00	79.5	446	1 288	2164	405	—
7500	1800	1.120	1.447	2605	4 00	81 5	464	1 288	2318	426	—
8000	1920	1 120	1.426	2738	4.00	83 0	481	1 288	2473	446	—
8500	2040	1 120	1.409	2874	4 00	85 0	497	1 288	2628	467	—
9000	2160	1 120	1.400	3024	4.00	87 0	515	1 288	2782	486	—
9500	2280	1 120	1 400	3192	4 00	89 0	535	1 288	2937	504	—
10000	2400	1 120	1 400	3360	4 00	91 5	555	1 288	3091	522	—
11000	2640	1 120	1.400	3696	4 00	95 0	595	1 288	3400	560	—
12000	2880	1 120	1 400	4032	4 00	99 0	634	1 288	3709	596	—
13000	3120	1 120	1 400	4368	4 00	103 0	673	1 288	4019	633	—
14000	3360	1 120	1 400	4704	4 00	106 5	712	1.288	4328	668	—
15000	3600	1.120	1 400	5040	4 00	109 5	751	1.288	4637	704	—
16000	3840	1 120	1 400	5376	4 00	112.5	789	1.288	4946	740	—
17000	4080	1 120	1 400	5712	4.00	115 5	827	1 288	5255	776	—
18000	4320	1 120	1 400	6048	4 00	118.0	863	1.288	5564	810	—
19000	4560	1 120	1 400	6384	4 00	120 0	899	1.288	5873	844	—
20000	4800	1 120	1 400	6720	4.00	120.0	900	1.288	6182	877	—

[a]Includes pickup allowance and correction for difference between test and operating conditions.

and grate is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired to the calorific value of 1 lb of fuel as fired.

2. *Liquid and Gaseous Fuels.* The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound or cubic foot of fuel to the calorific value of 1 lb or cubic foot of fuel respectively.

The following efficiencies apply to current designs of boilers operated under favorable conditions at their gross output ratings. Some older boilers designed primarily for hand firing may have lower efficiencies when automatically fired.

Anthracite, hand fired	60 to 75 per cent
Bituminous Coal, hand fired	50 to 65 per cent
Stoker fired	60 to 75 per cent
Oil and Gas fired	70 to 80 per cent

Higher efficiencies for hand fired bituminous coal may be obtained by careful firing of either a regular or a smokeless boiler.

RATING OF BOILERS

In referring to boiler rating it is necessary to know the basis on which the rating has been established in order to understand the exact meaning of the term. The following example will illustrate the meaning of three ratings which might be established for the same boiler.

Assume that an installation has the following loads determined in accordance with the section *Selection of Boilers:*

Net Load	1000 sq ft of steam radiation
Piping Tax	200 sq ft of steam radiation
Design Load	1200 sq ft of steam radiation
Pickup Allowance	240 sq ft of steam radiation
Maximum or Gross Load	1440 sq ft of steam radiation

A boiler that is just large enough to carry this system might be said to have a ***net load rating*** of 1000 sq ft, a ***design load*** rating of 1200 sq ft, or a ***gross load*** rating of 1440 sq ft, depending on the basis on which the boiler is rated.

On a *net load* basis the boiler would be rated 1000 sq ft of steam radiation and would have sufficient excess capacity to supply the *normal piping* and *pickup load.* Net $I=B=R$ Ratings, *SBI* Net Ratings, and Net Load Ratings of the *Heating, Piping and Air Conditioning Contractors National Association* are established on this basis.

On a *design load* basis the boiler would be rated 1200 sq ft of steam radiation and would have sufficient excess capacity to supply the *pickup load.* It would be of adequate size for a system in which the *sum* of the *net load* and the *piping heat loss* did not exceed 1200 sq ft of steam radiation. The *SBI* Ratings shown in columns 1, 2, 3, 10, 11 and 12 of Table 2 (not to be confused with *SBI* Net Rating) are established on a *design load* basis.

On a *gross output* basis of rating the boiler would be rated 1440 sq ft of steam radiation and would be of adequate size for a system in which the sum of the *net load, piping load,* and *pickup load* did not exceed 1440 sq ft of steam radiation. Gross $I=B=R$ Output and *A.G.A.* Ratings are established on a *gross output* basis.

In the determination of boiler ratings, the *Gross Output* is the quantity of heat available at the boiler nozzle with the boiler normally insulated

and when operating under limitations stipulated in the code or method by which the boiler is rated. The boiler may be capable of producing a greater nozzle output but in doing so would exceed some of these limitations.

SELECTION OF BOILERS

Factors Involved in Boiler Selection

The *Maximum Load* or *Gross Load* on the boiler is the sum of the four following items.

The *Design Load* is the sum of items 1, 2, and 3.

The *Net Load* is the sum of items 1 and 2.

1. *Radiation Load.* The estimated heat emission in Btu per hour of the connected radiation (direct, indirect, or forced convection coils) to be installed.

The connected radiation is determined by calculating the heat losses for each room in accordance with data given in Chapters 6, 8, and 14. The sum of the calculated heat losses for all the rooms represents the total required heat emission of the connected radiation expressed in Btu per hour. As practically all boilers are now rated on a Btu basis, it is unnecessary to convert the radiation load to equivalent square feet of equivalent radiation.

2. *Hot Water Supply Load.* The estimated maximum heat in Btu per hour required to heat water for domestic use.

When the hot water supply is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. A common practice is to add 240 Btu per hour to the radiation load for each gallon of storage tank capacity. For more specific information see Chapter 50.

3. *Piping Tax.* The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

As the heating industry as a whole is not entirely agreed upon piping tax allowances for different sizes of installations it is better to compute the heat emission from both bare and covered pipe surface in accordance with data in Chapter 28. In average house heating systems, it is common practice to consider the piping tax to be equal to 25 per cent of the Net Load. In determining Net $I=B=R$ Ratings from Gross $I=B=R$ Output, the piping factor allowed varies from 30 per cent for small boilers to 12 per cent for larger boilers.

4. *Warming-Up or Pick-Up Allowance.* The estimated increase in the normal load in Btu per hour caused by the heating up of the cold system.

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature, and heating up cold radiation and piping. The factors to be used for determining the allowance to be made should be selected from Table 4.

Other items to be considered in boiler selection are:

(*a*) Efficiency with hard or soft coal, gas, or oil firing, as the case may be.

(*b*) Grate area with hand fired coal, or fuel burning rate with stokers, oil, or gas.

(*c*) Combustion space in the furnace.

(*d*) Type of heat liberation, whether continuous or intermittent, or a combination of both.

(*e*) Convenience in firing and cleaning.

(*f*) Adaptability to changes in fuel and kind of attention.

(*g*) Height of water line.

(*h*) Miscellaneous items such as draft available, possibility of future extension, possibility of break-down, and head room in the boiler room.

(*i*) The most economical size of boiler is usually one that is just the right size for the load. Either larger or smaller boilers may be less economical.

Selection of Cast-Iron Boilers

Net load ratings of cast-iron boilers are usually available from manufacturers' catalogs. They may also be obtained conveniently from

published tables of $I=B=R$ ratings[6], or from recommendations of the *Heating, Piping and Air Conditioning Contractors National Association*[7] and can be used in selection of boilers, unless the heating system contains an unusual amount of bare pipe, or the nature of the connected load is such that the normal allowances for pipe loss and pickup do not apply. In such a case, the selection must be based on the gross output.

Selection of Steel Heating Boilers

SBI catalog ratings in accordance with the previously mentioned *Steel Boiler Institute, Inc.* code are intended to correspond with the estimated *design load*. When the heat emission of the piping is not known, the net load to be considered for the boiler may be determined from Tables 1 and 2. The difference between design load and net load represents an amount which is considered normal for piping loss of the ordinary heating system.

Boilers with less than 177 sq ft of heating surface and having *SBI* net ratings (steam) of not more than 3,000 sq ft if mechanically fired and 2,480 sq ft if hand fired, are classified as residence size. An insulated residence boiler for oil, gas, or stoker firing may carry a *net load* expressed in square feet of steam radiation of not more than 17 times the square feet of heating surface in the boiler, provided the boiler has been tested in accordance with the *SBI* Code for Testing Oil-Fired Steel Boilers at output rates of 125, 150, and 175 per cent of the *SBI* Net Rating. The *SBI* Net Rating (square feet steam) for hand-fired residence boilers is not greater than 14 times the heating surface. If the heat loss from the piping system exceeds 20 per cent of the installed radiation, the excess is to be considered as a part of the *net load*.

Selection Based on Heating Surface and Grate Area

Where neither the *net load* nor *gross output* ratings based upon performance tests are available, a good general rule for conventionally designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load. This is equivalent to allowing 10 sq ft of boiler heating surface per boiler horsepower. In this case it is assumed that the *maximum load* including the warming-up allowance will be provided for by

TABLE 4. WARMING-UP ALLOWANCES FOR HAND-FIRED LOW-PRESSURE STEAM AND HOT WATER HEATING BOILERS[a,b,c]

DESIGN LOAD (REPRESENTING SUMMATION OF ITEMS 1, 2, AND 3)		PERCENTAGE CAPACITY TO ADD FOR WARMING-UP[c]
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation.

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C. Taggart (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 292); Report of A.S.H.V.E. Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 35); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c]This table refers to *hand-fired, solid fuel boilers*. A factor of 20 per cent over design load is adequate when *automatically-fired fuels* are used.

[d]240 Btu per square foot.

operating the boiler in excess of the *design load,* that is, in excess of the 100 per cent rating on a boiler-horsepower basis. *SBI* ratings for hand firing are based on 10 sq ft of heating surface per boiler horsepower.

Due to the wide variation which may be encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{1}$$

where

G = grate area, square feet.
H = required *gross output* of the boiler, Btu per hour (see Selection of Boilers).
C = desirable combustion rate for fuel selected, pounds of dry coal per square foot of grate per hour (see Table 5).
F = calorific value of fuel, Btu per pound.
E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required *gross output* of the boiler of 500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 per cent.

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Equation 1. With small boilers, where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 per cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

TABLE 5. PRACTICAL COMBUSTION RATES FOR COAL-FIRED HEATING BOILERS OPERATING AT MAXIMUM LOAD ON NATURAL DRAFT OF FROM 1/8 IN. TO 1/2 IN. WATER[a]

KIND OF COAL	SQ FT GRATE	LB OF COAL PER SQ FT GRATE PER HOUR
No. 1 Buckwheat Anthracite	Up to 4 5 to 9 10 to 14 15 to 19 20 to 25	3 3½ 4 4½ 5
Anthracite Pea	Up to 9 10 to 19 20 to 25	5 5½ 6
Anthracite Nut and Larger	Up to 4 5 to 9 10 to 14 15 to 19 20 to 25	8 9 10 11 13
Bituminous	Up to 4 5 to 14 15 and above	9.5 12 15.5

[a]Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

Selection of Gas-Fired Boilers

After determining the *net load* for the installation, gas designed boilers can usually be selected from manufacturers' tables of *net load ratings* which are based on piping and pickup allowances varying from 56 per cent for boilers of 200 sq ft and less to 35 per cent for boilers of 4000 sq ft and larger. If the piping and pickup load or other factors create an unusual load, a boiler should be selected which has an *A.G.A. output rating* equal to the maximum output required. Detailed recommendations for selection of gas designed boilers are given in the *A.G.A.* publication, Comfort Heating[8].

SPACE LIMITATIONS

Boiler rooms should, if possible, be situated at a central point with respect to the building and should be designed for a maximum of natural light. The space in front of the boilers should be sufficient for firing, stoking, ash removal and cleaning or renewal of flue tubes, and should be at least 3 ft greater than the length of the tubes.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various dampers, cleanouts, and trimmings. The space at the rear of the boiler should be ample for the chimney connection and for cleanouts. With large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories and for proper installation of piping. In general the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating, especially, the height above the boiler water line is of vital importance.

When steel boilers are used, space should be provided for the removal and replacement of tubes.

CONNECTIONS AND FITTINGS

Steam outlet connections should be the full size of the manufacturers' tappings in order to keep the velocity of flow through the outlet reasonably low and to avoid fluctuation of the water line and undue entrainment of moisture, and should extend vertically to the maximum height available above the boiler. A steam velocity in boiler outlets not exceeding 25 to 30 fps at maximum load is recommended unless data are available to show that a higher velocity is satisfactory.

Particular attention should be given to *fitting connections* to secure conformity with the *ASME* Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections, and safety valve capacity.

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This facilitates cleaning sludge from the bottom of the boiler sections through the large plugged openings. An equivalent clean-out plug should be provided in the case of a single return connection.

Blow-off or drain connections should be made near the boiler and so arranged that the entire system may be drained of water by opening the

drain cock. In the case of two or more boilers separate blow-off connections must be provided for each boiler on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

For further data on pipe connections for steam and hot water heating systems, see Chapter 23 and 24 and the *ASME* Boiler Construction Code for Low Pressure Heating Boilers.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be air-tight and as short and direct as possible, preference being given to long radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining and where practicable it should be grouted from the inside of the chimney. A thimble or sleeve usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer should always be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement which cannot be reached after assembly is complete must be finished as the assembly progresses.

Five precautions that should be taken in all installations to prevent damage to the boiler are:

1. There should be provided proper and convenient drainage connections for use if the boiler is not in operation during freezing weather.

2. Strains on the boiler due to movement of piping during expansion should be prevented by suitable anchoring of piping and by proper provision for pipe expansion and contraction.

3. Direct impingement of too intense local heat upon any part of the boiler surface. as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material.

4. Condensation in steam systems must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.

5. Automatic boiler feeders and low water cut-off devices which shut off the source of heat if the water in the boiler falls below a safe level are recommended for mechanically fired boilers.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing; (*d*) boiler too small; (*e*) improper piping; (*f*) improper arrangement of sections; (*g*) heating surfaces covered with soot;

(*h*) insufficient radiation installed; and (*i*) with mechanical firing, fuel burning equipment too small.

2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; and (*c*) boiler operating at excessive output.

3. *Water disappears from gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and return piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; and (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area; (*d*) excessive rate of output; and (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers.* This may be due to: (*a*) poor draft resulting from air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; and (*e*) boiler too small for the load

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion; (*c*) too low a rate of combustion; and (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; and (*e*) improper reduction in breeching size.

8. *Low carbon dioxide.* This may be due on oil burning boilers to: (*a*) improper adjustment of the burner; (*b*) leakage through the boiler setting; (*c*) improper fire caused by a fouled nozzle; or (*d*) to an insufficient quantity of oil being burned.

Cleaning Boilers

All boilers are provided with flue clean-out openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom of the boiler and form sludge. These impurities tend to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a surface blow connection of at least 1¼ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure, and while fire is burning briskly open valve in blow-off line. When pressure recedes, close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb, close blow-off, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service. The use of soda, or any alkali, vinegar or any acid is not recommended for cleaning heating boilers because of the difficulty of complete removal and the possibility of subsequent injury, after the cleaning process has been completed.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use, as given by the boiler manufacturer, should be carefully followed.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulfur in the soot with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.

2. All machined surfaces should be coated with oil or grease.

3. Connections to the chimney should be cleaned and in case of small boilers the pipe should be placed in a dry place after cleaning.

4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard that some one may inadvertently build a fire in a dry boiler, however, it is safer to keep the boiler filled with water, particularly in residential installations Air can be excluded from a steam boiler by raising the water level into the steam outlets. A hot water system usually is left filled to the expansion tank.

5. The grates and ashpit should be cleaned.

6. Clean and repack the gage glass if necessary.

7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.

8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings, and regulator parts.

WARM AIR FURNACES

Warm air heating furnaces of a number of types and a wide range of sizes are listed and illustrated in the *Catalog Data Section.*

TYPES OF FURNACES

Warm air furnaces may be classified in several different ways:

a. According to method of heat distribution—these are either gravity or mechanical (blower) furnaces.

b. According to fuels for which the furnaces are designed—these are coal hand-fired or stoker-fired, oil, gas, or wood.

c. According to materials of construction—they are cast-iron, low carbon steel, and occasionally high temperature steel alloys.

d. According to design or construction, such as drum and radiator, tubular, horizontal, etc.

Gravity Warm Air Furnaces

A gravity furnace is one in which the motive head producing air flow depends upon the difference in density between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing. Since this gravity head is relatively low, the furnace must have low internal resistance to the flow of air and relatively large areas must be available for free circulation within the furnace casing. It is common practice to provide approximately 50 per cent free air area through gravity type furnaces.

Furnaces for gravity type systems are available in designs suitable for central heating, pipeless furnace, or unit floor furnace installations. Booster fans are sometimes used in conjunction with gravity design

systems, to increase air circulation. Where a fan is to be used with a furnace casing sized for gravity air flow, some form of baffling must be employed to restrict the free area within the casing and to force impingement of the air against the heating surfaces. Where square casings are used, the corners must be baffled.

Mechanical Warm Air Furnaces

Mechanical or forced warm air furnaces include fans or blowers as integral parts for the purpose of circulating the air and usually include air filters.

Fans and Motors

Centrifugal fans with either backward or forward curved blades are the type most commonly used. Motors may be mounted on the fan shaft or connected to the fan by a belt drive. Adjustable pulleys are desirable to provide means of regulating the quantity of air distributed to the heated spaces. Either the motor load or the noise may limit the maximum operating fan speed. Two-speed motors have given successful operating results and are recommended. Motors and mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with the fan housing.

Filters

Several types of filters are available for mechanical warm air furnace applications and are discussed in Chapter 33. For maximum efficiency and life under operating conditions, filters should not be subjected to a temperature in excess of 150 F. Filters should have at least 80 per cent average efficiency on an 8-hr test at a maximum resistance of 0.25 in. of water. Filter resistance rises rapidly with the accumulation of dirt, and may reduce the air circulation over heating surfaces. In domestic furnaces, the maximum velocity, based on nominal filter area, should not exceed 300 fpm.

Fuel Utilization

A combustion rate of from 5 to 8 lb of coal per (square foot of grate) (hour) is recommended for residential furnaces. A higher combustion rate is permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes the ratio may be as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of a number of furnaces, using one or more fans.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. Furnaces for burning oil fuel are usually designed for blow-through installations so that the pressure in the air space is higher than that in the combustion chamber or flues. The *National Warm Air Heating and Air Conditioning Association* has prepared a Tentative Code for Testing and Rating of Oil-Fired Furnaces. Compact fan-furnace-burner units are available, suitable for basement, closet, or even attic installations.

Gas-fired forced air furnaces should conform in construction and performance to *A.G.A.* Approval Requirements.

Heavy Duty Fan Furnaces

Fan furnaces for large commercial and industrial buildings, churches. schools, etc., are available in sizes ranging from 300,000 to 3,000,000 Btu per (hour) (unit). Heavy duty furnace heaters may be arranged in battery combinations of one or more units.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu per hour and also in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area of furnaces for this type of work should never be less than 30 to 1 and as indicated previously may run as high as 50 to 1.

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 43. Ducts are designed by the method outlined in Chapter 41.

MATERIALS AND CONSTRUCTION

Cast-Iron Furnaces

Cast-iron furnaces are made in a multiplicity of designs or shapes. For solid fuels they are generally of round sectional construction, the sections being cemented or bolted together. Various types of radiators for secondary convection heat transfer are employed. Such radiators are of the circular, doughnut type, or tubular type.

Cast-iron is frequently used in the construction of gas or oil-fired furnaces, designs varying considerably with two general types in common use: multi-sectional type and those with single combustion chambers having auxiliary secondary surface.

Cast-iron furnaces are made in capacities ranging from those for small insulated residence application with inputs of 40,000 Btu per hour or less, to capacities as large as 600,000 Btu per hour.

Cast-iron furnaces are usually constructed with a minimum sectional thickness of ¼ in. and effectively resist high temperatures and corrosion. They usually have a fairly large heat capacity because of their mass, which provides a distinct *fly wheel* or carry-over heating effect.

Steel Furnaces

Formed sheet steel construction is frequently used in furnace design. Welding, riveting, or both are used to join the formed metal. The use of steel castings, however, is rare, because of the cost, and because high stresses are not encountered in normal furnace construction. Types of design employed vary greatly, although perhaps the most common type consists of a drum and circumferential or rear radiator. Steel gas furnaces may also be sectional in design or may be combinations of common combustion chambers and sectional or tubular radiation surfaces connected to a flue gas collector.

Steel furnaces are made in capacities ranging from those for small insulated residence application with inputs of 40,000 Btu per hour to

capacities as large as 600,000 Btu. Steel furnaces have low heat capacities as a result of their relatively low mass and, therefore, deliver heat rapidly on demand.

RATING OF FURNACES—TESTING AND RATING CODES

Warm air furnaces are generally rated in Btu per hour output at the bonnet (point of heat generation) or at the register (point of heat delivery).

Rating Equations for Gravity Warm Air Furnaces[9]

Until a method of testing and rating gravity warm air furnaces has been developed, the following empirical rating equations are recommended by the *National Warm Air Heating and Air Conditioning Association.*

Gravity warm-air furnaces of conventional design, having ratios (of heating surface to grate area) of 15 to 1 or greater, and having a ratio of casing area to face area not less than 0.4, are rated by the following equations:

a. *Hand-fired furnaces Converted to Stoker, Gas, or Oil Firing.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times S \times 1.333 \qquad (2)$$

b. *Hand-fired furnaces, with ratios of heating surface to grate area greater than 15 to 1 and less than 25 to 1.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times S \times 1.333 \qquad (3)$$

c. *Hand-fired furnaces with ratios of heating surface to grate area in excess of 25 to 1.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times 25 \times G \times 1\ 333 \qquad (4)$$

where

S = heating surface, in square feet.
G = actual grate area, in square feet.

The *Register Delivery Rating* is equal to 0.75 x (*Bonnet Capacity*). The *Leader Pipe Rating* in square inches, formerly used as a rating unit, may be found by dividing the *Register Delivery Rating* by 136.

Heating Surface of Furnace

Prime heating surface is defined as surface above the top of the grate having *hot gases or live fuel on one side and circulating air over the other*, and in all cases is measured on the exterior or air side. The areas of the outer casing, the inner liner, and any radiation shields shall not be considered as heating surface.

In determining the amount of heating surface, extended surfaces are considered to be prime heating surface subject to the following limitations:

a. Extended heating surface may consist of fins, ribs, webs, lugs, or other projections from the prime heating surface. Projections less than ¼ in. thick at the base and extending more than 1 in. from the prime surface are classified as fins.

b. Integral fins are continuously welded to, or cast as a part of, the prime heating surface. Both sides are included as heating surface, subject to the following allowances:

Distance from Prime Surface	1st inch	2nd inch	3rd inch	Over 3 in.
Ratio of Effective Area to Total Area	0.40	0.30	0.20	None

c. Non-integral fins are spot welded to, or otherwise held in line contact with the prime heating surface. Both sides are included as heating surface, subject to the following allowances:

Distance from Prime Surface	1st inch	2nd inch	3rd inch	Over 3 in.
Ratio of Effective Area to Total Area	0 30	0.20	0.15	None

d. In the case of ribs, webs, or lugs more than ¼ in. thick at the base and extending less than 1 in. from the prime surface, the entire surface in contact with circulating air is included as heating surface.

e. In the case of ribs, webs, or lugs more than ¼ in. thick at the base and extending more than 1 in. from the prime heating surface the areas of both sides of the first inch are included as prime heating surface. The portions projecting beyond 1 in. are treated as integral fins.

Grate Area

Grate area is defined and treated for purpose of rating as follows:

a. The *nominal grate area* is defined as the total cross-sectional area of the bottom of the firepot. In steel furnaces the nominal grate area is the cross-sectional area inside the firebrick lining.

b. The *actual grate area*, used for calculating the ratios of heating surface to grate area, is the nominal grate area minus certain areas that cannot be considered as part of the grate itself. The following rules govern these deductions: (1) If a solid, continuous ledge extends around the grate and inside the firepot, any area of this ledge extending inside of a circle, the diameter of which is 1 in. less than the diameter of the bottom of the firepot, shall be deducted. (2) If separate, solid projections extend from the firepot towards the grate, the areas of any portions of these projections extending inside of a circle, the diameter of which is 3 in. less than the diameter of the bottom of the firepot, shall be deducted. (3) In the case of grates which are inclined, or are conical, the projected area is the same as the nominal grate area. The latter should, therefore, be used after making any necessary deductions.

Ratings for Forced Air Furnaces

For solid fuel burning, forced air furnaces having bonnet capacities between 80,000 and 250,000 Btu per hour, no standard method of test has been accepted, although eventually such codes will be developed. The *National Warm Air Heating and Air Conditioning Association* recommends empirical equations similar to Equations 2, 3, and 4 for gravity furnaces, except that a constant of 2265 is used in place of the 1785.

The following testing and rating codes have been generally accepted in the industry:

Commercial Standards CS-109-44 for rating solid fuel-burning, forced-air furnaces having bonnet outputs of 80,000 Btu per hour or less. This provides a method of rating small coal-fired forced-air furnaces by test.

A Tentative Code for Testing Oil-Fired Furnaces. This code has been adopted by the *National Warm Air Heating and Air Conditioning Association* for rating oil-fired furnaces by test.

The American Gas Association method of rating gas-fired furnaces based upon performance under tests. This is described in the Approval Requirements for Central Heating Gas Appliances.

Commercial Standards 113-44 is a method of rating oil-burning floor furnaces by test.

Commercial Standard CS 104-43 is a method of rating warm air furnaces equipped with pot-type oil burners by test.

Various codes covering the construction and performance of appliances as related to fire hazards have been developed by Underwriter Laboratories, Inc. In addition, there

are many municipal codes [10] which regulate construction and installation of furnace equipment.

The Yardstick of the *National Warm Air Heating and Air Conditioning Association* provides criteria for evaluating a furnace design and installation against industry accepted standards.

FURNACE EFFICIENCY

Rating formulae of the *National Warm Air Heating and Air Conditioning Association* are based on 55 per cent efficiency for gravity coal furnaces and 65 per cent efficiency for forced air coal furnaces. In the tentative Oil Testing Code the contemplated minimum efficiency is 70 per cent for oil fired forced air furnaces. Gravity gas furnaces approved by the *American Gas Association* are assigned a rating based on 75 per cent efficiency. All *forced air gas-fired furnaces* approved by *American Gas Association* are assigned a rating based on 80 per cent efficiency.

DESIGN CONSIDERATIONS

Considerations of prime importance in the design of warm air furnaces and some general suggestions to be observed in connection with each are as follows:

1. *Adequate heat transfer surface.*

   a. Heat transfer rates of 2,000 to 4,500 Btu per (hour) (square foot) of heating surface may be obtained without unduly high metal temperatures.

   b. Fins, pins and bosses are frequently used to add surface and to break down superficial gas films, both on gas-to-metal and metal-to-air surfaces.

   c. Surface and stack (flue gas) temperatures are good indications of the amount and effectiveness of the heating surfaces.

2. *Safe and efficient combustion of fuel.*

   a. Proper mixture of fuel and air is necessary for efficient combustion. This necessitates careful attention to the design of grates, nozzles, burners, air inlet areas and location, and combustion chamber baffling.

   b. Regulation of the quantity and the distribution of the air for combustion should be provided by use of check dampers, draft regulators, draft hoods, air shutters and air orifices.

   c. Total draft loss through appliances should not exceed that available from chimneys which would normally be obtainable in the size of building which the appliance will supply with heat.

   d. The use of ignition safety devices such as safety pilots, hold-fire controls, and the like is recommended.

3. *Fuel Capacity of Appliance.*

   a. With solid fuels adequate coal capacity should be provided for at least 5 hr of operation at the maximum rated combustion rate.

4. *Adequate circulation of air over heating surface.*

   a. In gravity furnaces, free air space between casing and heat exchanger should be great enough to permit free flow over all surfaces.

   b. Forced air furnace design must include fans having proper capacity and suitable performance characteristics. Internal static pressures must be minimized without losing the advantages of high velocity circulation over the heat exchanger surfaces.

   c. The air flow over the heating surface must be directed to obtain maximum efficiency and to eliminate hot spots and air noises.

   d. Air velocities at bonnet should not be much in excess of 1,000 fpm and air temperature distribution at the furnace outlet should be uniform within approximately $\pm$ 30 deg.

square feet (steam) on the basis of heating surface with limitations set for grate area, furnace volume, and furnace height. The residential boilers (defined as those having not more than 177 sq ft of heating surface) are rated on tests for oil-fired boilers, with limitations in relation to heating surface and testing conditions. Stoker-fired and gas-fired *residential* boilers are rated (*SBI* Net Rating) not in excess of the oil-fired rating. Hand-fired residential boilers are rated (*SBI* Net Rating) not greater than 14 times the heating surface.

Tables 1 and 2 show the *SBI* ratings of residential and commercial steel boilers respectively.

The *Institute of Boiler and Radiator Manufacturers* has adopted a Code [5] for rating cast-iron heating boilers based upon performance obtained under controlled test conditions. This Code applies to all sectional cast-iron heating boilers except those of magazine feed type.

The *Gross I=B=R* Output is obtained by test and is subject to certain limiting factors. For hand fired boilers, the number of boilers of a series to be tested, the minimum over-all efficiency, the minimum time limit (the time an *Available Fuel Charge* will last when burned at a rate which will produce the *Gross I=B=R Output*), the chimney area and height, and the draft in the stack are all subject to the limits established in the Code. Tests are run using anthracite coal of standard specification. Bituminous coal and coke ratings are the same as for anthracite coal.

For automatically fired boilers, the number of boilers of a series to be tested, the flue gas temperature and analysis, the minimum over-all efficiency, the draft loss through the boiler, and the heat release in the combustion chamber are subject to limitation by the Code [5]. Automatically fired boiler ratings are established by oil fired tests using gun type oil burners and commercial grade No. 2 fuel oil. Stoker fired and gas fired ratings (where no *A.G.A.* Rating is published) are based on the *Gross I=B=R Output* obtained by oil fired tests.

The *Net I=B=R Rating* is determined from the *Gross I=B=R Output* by applying specified *Piping and Pickup Factors* which range from 2.36 to 1.40 for hand fired boilers and from 1.56 to 1.288 for automatically fired boilers. In both cases, the factor decreases as the boiler size increases. Table 3 is abstracted from the *I=B=R Rating Tables* in the Code and illustrates the relationship between *Net I=B=R Rating* and *Gross I=B=R Output.*

The *American Gas Association* has adopted a method of rating gas designed boilers based upon performance under tests. This is described in Approval Requirements for Central Heating Gas Appliances.

The *Heating, Piping and Air Conditioning Contractors National Association* has adopted a method of rating boilers based on their physical characteristics for those boilers that are not rated in accordance with the *SBI* or *I=B=R* Codes. Ratings are expressed on a Net Load basis in square feet of steam radiation.

BOILER EFFICIENCY

The term *efficiency* as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace*

1000 Btu per pound are required. There is a trend in present practice toward heating the water in addition to heating the air. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device.

Sprays are usually controlled by solenoid valves wired in parallel with the fan motor. The water supply may, in turn, be controlled by a humidity-controlling device located in one of the living rooms, so that the washer will operate at all times when the fan is in operation, unless the relative humidity should rise beyond a desirable percentage. Sprays used in connection with commercial or heavy duty plants should be a regulation type of commercial spray. In all cases provision must be made to flush out accumulation of lime and dirt.

REFERENCES

[1]—See A.S H V E. TRANSACTIONS, Vol. 35, 1929, pp. 322 and 332.

[2]—See A.S.H V.E. TRANSACTIONS, Vol. 36, 1930, p. 42.

[3]—See A.S.H.V.E TRANSACTIONS, Vol. 37, 1931, p. 23.

[4]—See A.S H.V.E. TRANSACTIONS, Vol. 44, 1938, p 366

[5]—*I* = *B* = *R* Testing and Rating Code for Low Pressure Heating Boilers (*Institute of Boiler and Radiator Manufacturers*).

[6]—*I* = *B* = *R* Ratings for Cast-Iron Boilers (*Institute of Boiler and Radiator Manufacturers*)

[7]—Engineering Standards, Part II, Net Square Feet Radiation Loads in 70 Deg Fahr, Recommended for Low Pressure Heating Boilers, 1943 (*Heating, Piping and Air Conditioning Contractors National Association*)

[8]—Comfort Heating, 1938, pp 35 to 39 (*American Gas Association*)

[9]—For definition of Heating Surfaces and Ratings of Wood-Burning Furnaces see Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems, (*National Warm Air Heating and Air Conditioning Association*).

[10]—Recommended forms for municipal installation and fire codes are included in Volume 7—Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems (*National Warm Air Heating and Air Conditioning Association*).

CHAPTER 19

...eys and Draft Calculations

...nd Available Draft, Determining Chimney Sizes, ...ting Required Draft, Domestic Chimneys, Ob- ...Performance, Draft Requirements of Domestic ...Chimneys for Gas Heating, Construction Details, General Considerations

...e older sense, is a current of air and the draft of a ...r is that air current which flows through the fire-box ...xygen for combustion. In engineering, however, the ...e to mean that pressure difference which causes this ...nd the word will be used in this sense in this chapter.

...neasured in inches of water and it is proper to speak ...ire-box or in the smoke breeching, etc., meaning the ...e between the gases within and the air without those ...Draft is called positive when the pressure within such ...hat outside.

... as natural and mechanical, depending on whether it ...mney or by a blower, and mechanical draft is further ...l or forced, depending on whether the air is drawn ...hrough the combustion chamber.

...ve both to create a draft and to dispose of combustion ...ble height. For the latter purpose, chimneys, stacks, ...ps, funnels, are used in conjunction with mechanical-

THEORETICAL DRAFT

... two equal chimneys is heated while that in the other ...e heated chimney will be less heavy than that in the ... a manometer or other pressure gage connecting the ...will indicate a pressure difference, called natural draft. ... air at the tops of the two chimneys will be equal, so ...ference between them at the bottom will depend only ...the difference in density of the air they contain. The ...either chimney is inversely proportional to its absolute ...at the difference in pressure between them at the ...ortional to their height and to the difference between ...e absolute temperatures within them.

...he bottom of an unheated (and uncooled) chimney ...that of the air outside, so that the unheated chimney ...n the foregoing illustration. The manometer reading ...ts free connection is left open to the atmosphere.

...ons in conjunction with those of barometric pressure ...n density of flue gases from that of air lead to the

$$D_t = 2.96\ HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) \qquad (1)$$

...ney, feet.

...etric pressure, inches of mercury.

...at 0 F and 1 atmosphere pressure, pounds per cubic foot.

... gas at 0 F and 1 atmosphere pressure, pounds per cubic foot.

T_o = temperature of air surrounding the chimney, Fahrenheit degrees absolute.
T_c = average or effective temperature of the gases in the chimney, Fahrenheit degrees absolute.

The quantity D_t, yielded by the formula, is the pressure difference between the gas inside and air outside of the chimney, in inches of water, when no flow occurs in the chimney. The quantity is variously known as the theoretical draft, the static draft or the computed draft. It is very useful in predicting and analyzing chimney performance, but it is seldom if ever attained in an actual chimney because of the friction incident to gas flow, wind effects, etc.

AVAILABLE DRAFT

The available draft, D_a, *for large chimneys and stacks* has been estimated with apparent satisfaction in the past by means of formulas which in effect deduct an estimated friction loss from a theoretical draft determined as in Equation 1. The friction loss can be estimated by means of one of the formulas available for ducts, such as the Fanning equation. This procedure results in formulas for the available draft as follows:

For a *cylindrical* stack:

$$D_a = 2.96\, HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.00126\, W^2 T_c f L}{D^5 B_o W_c} \tag{2}$$

and for a *rectangular* stack:

$$D_a = 2.96\, HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.000388\, W^2\, T_c f L\, (x + y)}{xy^3\, B_o\, W_c} \tag{3}$$

where
D_a = available draft, inches water gage.
H = height of chimney above grate, feet.
B_o = existing barometric pressure, inches of mercury.
W_o = density of air at 0 F, 1 atmosphere pressure.
W_c = density of flue gas at 0 F, 1 atmosphere pressure.
T_o = temperature of atmosphere, Fahrenheit degrees absolute.
T_c = temperature of flue gas, Fahrenheit degrees absolute.
W = flue gas flow rate, pounds per second.
f = coefficient of friction.
L = length of friction duct (= H approximately), feet.
D = minimum diameter of round chimney, feet.
x and y = length and width of cross-section of rectangular chimney, feet.

The following notes facilitate the use of Equations 2 and 3.

1. The *barometric pressure*, represented by B_o, is the actual pressure at the site of the chimney and not the pressure reduced to sea level datum.

In general, the barometric pressure decreases approximately 0.1 in. Hg per 100 ft increase in elevation.

2. The *unit weight of a cubic foot* of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_c = 0.131 CO_2 + 0.095 O_2 + 0.083 N_2 \tag{4}$$

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_c may be assumed at 0.09.

The density effect on the chimney gases due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltrations in the chimney proper is disregarded. Though water vapor content is not disclosed by Orsat analysis, its presence tends to reduce the actual weight per cubic foot of chimney gases.

3. The *atmospheric temperature* is the actual observed temperature of the outside air

at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* decreases from the breeching connection to the top of the stack. This drop in temperature depends upon the material and construction of the stack, its tightness or freedom from leaks, its area, its height, and the velocity of the gases through it. The same chimney will suffer different temperature losses depending upon the capacity under which it is working and the variable atmospheric conditions. No general equation covering all these variables has been suggested, but from observations on chimneys varying in diameter from 3 to 16 ft and in height from 100 to 250 ft Equation 5 was deduced [1]:

$$T_c = \frac{3.13\, T_1 \left[\left(\frac{H_b}{3}\right)^{0.96} - 1\right]}{H_b - 3} \tag{5}$$

where

T_1 = absolute temperature at the center of the connection from the breeching, Fahrenheit degrees.

H_b = the height of the stack above center line connection to breeching, feet.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the materials of construction becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and in general constant for all conditions of operation. For reasons of simplicity and convenience to the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

In important chimney design, especially when the construction or the materials are unusual, it is recommended that use be made of the Reynolds number [2] in determining the friction factor, f.

The following problem illustrates the use of Equation 2:

Example 1. Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, B_0 = 29.92 in. Hg; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft. The chimney discharges 100 lb of gases per second.

Substituting these values in Equation 2 and reducing:

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^5 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 1 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in Example 1. When the chimney is under static conditions and no gases are flowing, the available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available draft decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss due to friction is equal to the theoretical intensity. The point of maximum draft and zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 2 and then plotting the results in the manner shown in Fig. 1.

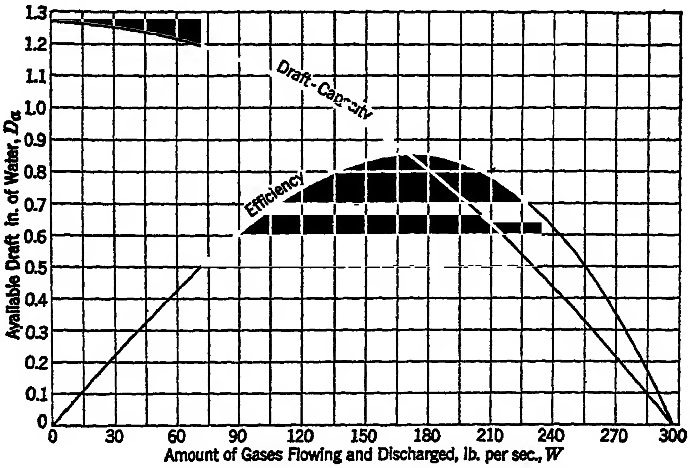


FIG. 1. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

Fig. 2 is a typical chimney performance chart giving the available draft for various gas flow rates and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cubic foot, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific conditions, a new chart may be prepared from Equation 2 or 3.

DETERMINING CHIMNEY SIZES

If the required performance for a proposed chimney is known and *if a chimney-gas velocity is assumed,* Equation 2 can be transposed to yield the necessary height and an equation can be developed for the required diameter. These operations result in the following equations:

$$H = \frac{D_T}{2.96B_O\left(\dfrac{W_O}{T_O} - \dfrac{W_C}{T_C}\right) - \dfrac{0.184fW_CB_OV^2}{T_CD}} \tag{6}$$

The weight of gas per second, $W = 12.075 \dfrac{D^2\,VB_O\,W_C}{T_C}$ from which

$$D = 0.288\sqrt{\frac{WT_C}{B_OW_CV}} \tag{7}$$

where

H = required height of chimney above grate, feet.
D = required minimum diameter of chimney, feet.
V = chimney gas velocity, feet per second.
D_T = total required draft, inches of water.

For large chimneys, it is usual to assume that total construction cost is least when the product HD (height × diameter) is minimum. On this assumption, the product of Equations 6 and 7 can be differentiated and

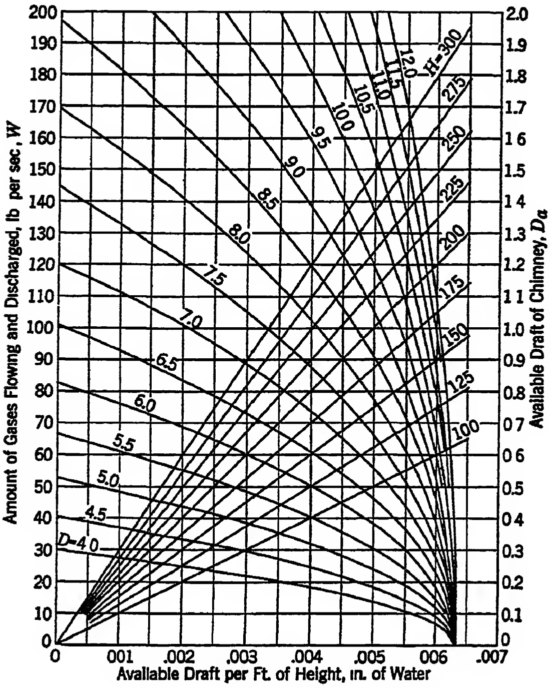


FIG. 2. CHIMNEY PERFORMANCE CHART

To solve a typical example: Proceed horizontally from a Weight Flow Rate point to intersection with diameter line; from this intersection follow vertically to chimney height line; from this intersection follow horizontally to the right to Available Draft scale. Starting from a point of Available Draft, take steps in reverse order.

the differential set equal to zero to find the minimum. Solution for velocity then yields the following equation:

$$V_e = \left(\frac{0.772T_c\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right)\sqrt{\frac{WT_c}{B_oW_c}}}{fW_c} \right)^{2/5} \quad (8)$$

where V_e = economical chimney gas velocity, feet per second.

Equations 6, 7 and 8 can of course be simplified if values are assumed for some of the factors in it. Some typical figures for boiler plants are:

Average chimney gas temperature 500 F	T_c = 960 F absolute
Average atmospheric temperature 62 F	T_o = 522 F absolute
Average coefficient of friction 0.016	f = 0.016
Average chimney, gas density, 0 F, 1 Atmosphere	W_c = 0.09 lb per cubic foot
Barometer reading, sea level	B_o = 29.92 in. Hg

When these values are substituted in Equations 8, 7 and 6 respectively, the results are:

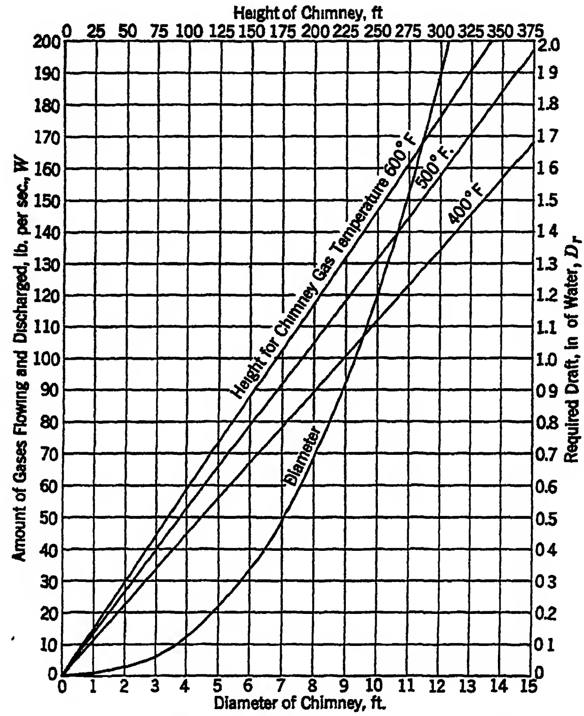
$V_e = 13.7W^{1/5}$ (9) $D = 1.5W^{2/5}$ (10) $H = 190D_r$ (11)

Fig. 3 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities computed from Equations 9, 10 and 11. They are based on the operating factors used in reducing Equations 6, 7 and 8 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only,

or where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 6, 7 and 8.

FACTORS AFFECTING REQUIRED DRAFT

The foregoing considerations deal with chimney size selection when the required draft and flue gas volume and temperature are known. The required draft is, of course, equal to the sum of all the resistances to gas flow from the ash pit door to and including the chimney connection.



Diameter values also for gas temperatures of 400, 500 and 600 F.

FIG. 3. ECONOMICAL CHIMNEY SIZES

Fig. 4 presents information on the fuel-bed draft loss for various kinds of coal burned at different rates and rough generalizations can be given for the losses in the flue passages of boiler or furnace, but, on account of the great differences in such devices, more reliable data on their flue gas volume temperature and flue resistance should be obtained for design purposes from their respective manufacturers.

Flue gases encounter resistance to flow in breechings or smoke pipes and this can probably be treated with sufficient accuracy by means of the method used for air ducts. (See Chapter 41.) The friction in straight ducts can be estimated by means of the last terms of Equations 2 and 3.

Also, the temperature of flue gases falls during passage through breechings or flue pipes. For uninsulated surfaces this probably can be adequately estimated by assuming a loss of heat from the flue gas of 3 Btu per

hour per square foot for each Fahrenheit degree temperature difference between the gases and surrounding air.

DOMESTIC CHIMNEYS

The height of a chimney for a residence or apartment is generally limited by the height of the building since it is desirable to have the chimney architecturally congruous with the building. The height desirable from an architectural viewpoint and the location of the chimney may be disadvantageous to the operation of the boiler or furnace and it is therefore important that the manufacturer of the fuel burning appliance to be installed be consulted in regard to the adequacy of the chimney.

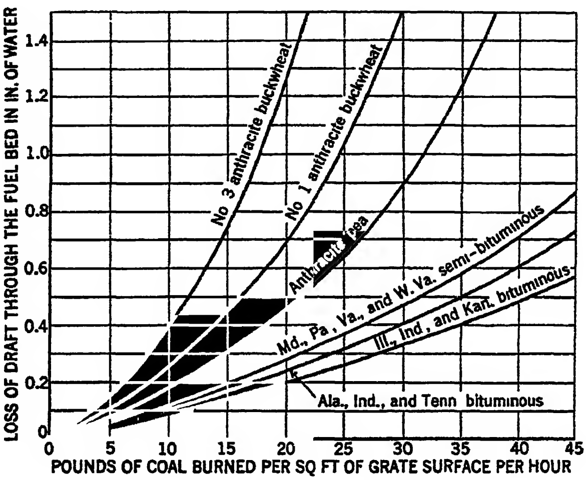


FIG. 4. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

A chimney in order to provide satisfactory performance must have adequate *height* and *area,* be of permanently tight construction and should be as smooth internally as practicable.

It should be remembered that mechanically fired devices, oil burners and stokers, are equipped with blowers so that, with these devices, the chimney is not required to overcome the resistance of a fuel bed. Nevertheless, a draft in the fire box, of about 0.03 in. of water is considered desirable so that any small openings in the fire box or flue passages will result in leakage of air inward, and not leakage of combustion products outward. This is not to be taken to condone leaks in fire boxes. Such leaks adversely affect plant efficiency.

OBSERVED TEST PERFORMANCE

The observed performances of some brick chimneys[3] are given in Tables 1 and 2.

The tests on which these data are based were made at various outside temperatures as shown and, to make them comparable among themselves, the observed drafts were corrected to 32 F, 1 atmosphere pressure, by the formula:

TABLE 1. TEMPERATURE AND DRAFT IN 9 IN. BY 9 IN. MASONRY CHIMNEY[a]

Chimney Height, Ft		Fuel Oil Equivalent[c] of Chimney Gases		Flue Gas Temp at Inlet, F	Average Chimney Gas Temp. F	Outside Temp. F	Observed Draft Inch Water	Computed Static Draft Inch Water	Corrected Draft[b]		Indicated Friction Loss Inch Water
Nominal	Height Above Thimble	Fuel Oil Gal/Hr	CO_2 Per Cent						Observed Inch Water	Computed Static Inch Water	
35	31.5	0.5	10	236	142	68	0.040	0.053	0.076	0.089	+0.013
35	31.5	1 5	8	200	158	81	0.043	0.055	0 086	0.098	+0.012
35	31.5	0 5	10	400	224	77	0 081	0.095	0 128	0 142	+0.014
35	31.5	1 5	8	400	278	80	0.092	0.117	0.139	0.164	+0.025
35	31.5	0.5	10	1000	488	77	0 163	0.191	0.206	0.234	+0.028
35	31.5	1 5	8	1000	601	78	0.169	0.214	0.216	0.261	+0.045
30	27.5	0 5	10	250	160	83	0.043	0.049	0.070	0.076	+0.006
30	27 5	1.5	8	250	195	84	0.055	0.064	0 097	0.106	+0.009
30	27.5	0 5	10	400	213	68	0.075	0.084	0.107	0.116	+0 009
30	27.5	1.5	8	400	269	72	0.094	0.108	0.125	0.139	+0.014
30	27.5	0.5	10	1000	460	64	0.146	0 169	0 174	0.197	+0.023
30	27 5	1.5	8	1000	608	72	0.168	0.193	0.204	0.228	+0.025
25	22.3	0.5	10	250	171	92	0.034	0.037	0.072	0.076	+0.003
25	22.3	1.5	8	200	171	94	0 034	0.036	0.074	0.076	+0.002
25	22.3	0.5	10	400	243	78	0.066	0 073	0.097	0.105	+0.006
25	22.3	1.5	8	400	301	79	0.085	0.091	0.116	0.123	+0.006
25	22.3	0 5	10	1000	504	79	0.124	0.137	0.154	0.167	+0.013
25	22.3	1.5	8	1000	642	80	0.146	0.159	0.177	0.190	+0.013
20	17 2	0.5	10	250	179	94	0 029	0.032	0.059	0 062	+0 003
20	17.2	1.5	8	200	188	94	0.031	0.032	0 059	0.060	+0.001
20	17.2	0.5	10	400	270	81	0.058	0.062	0.082	0.086	+0.004
20	17.2	1.5	8	400	297	55	0.075	0.081	0.087	0.093	+0.006
20	17.2	0.5	10	600	330	50	0.086	0.091	0.095	0.100	+0.005
20	17.2	1.5	8	600	425	55	0.104	0.106	0.116	0.118	+0.002
20	17.2	0.5	10	1000	526	37	0.120	0.130	0.123	0.134	+0 010
20	17.2	1.5	8	915	588	44	0.129	0.134	0.135	0.141	+0.005
15	11.3	0.5	10	250	182	87	0.023	0.023	0.041	0 041	+0.000
15	11.3	1 5	8	200	180	84	0 025	0.024	0.042	0 041	−0.001
15	11 3	0.5	10	400	258	32	0.053	0.055	0.054	0.056	+0.002
15	11.3	1.5	8	400	293	43	0 056	0.057	0 060	0 061	+0.001
15	11.3	0.5	10	600	369	34	0 068	0 071	0.069	0.071	+0.003
15	11 3	1 5	8	600	428	40	0 072	0.076	0 075	0.078	+0.004
15	11.3	0.5	10	1000	577	32	0.094	0.092	0.094	0.092	−0 002
15	11.3	1 5	8	1000	654	55	0.092	0.090	0.100	0.098	−0.002

[a]Actual inside dimensions of flue lining 7¼ × 7¼ in.
[b]Corrected for outside temperature 32 F. Barometer 29.92 in. Hg
[c]½ gal per hour of fuel oil burned with 10 per cent CO_2 produces 18.4 cfm or 1.38 lb per minute of flue gases (corrected to 70 F, 1 atmosphere pressure); 1½ gal per hour of fuel oil burned with 8 per cent CO_2 produces 67.5 cfm or 5.06 lb per minute of flue gas (corrected to 70 F, 1 atmosphere pressure).

TABLE 2. TEMPERATURE AND DRAFT IN 9 IN. BY 13 IN. MASONRY CHIMNEY[a]

Chimney Height, Ft		Fuel Oil Equivalent[c] of Chimney Gases		Flue Gas Temp. at Inlet, F	Average Chimney Gas Temp. F	Outside Temp. F	Observed Draft Inch Water	Computed Static Draft Inch Water	Corrected Draft[b]		Indicated Friction Loss Inch Water
Nominal	Height Above Thimble	Fuel Oil Gal/Hr	CO_2 Per Cent						Observed Inch Water	Computed Static Inch Water	
35	31.5	0.5	10	204	125	78	0.035	0.034	0.078	0.077	−0.001
35	31.5	1.5	8	206	155	82	0.047	0.052	0.093	0.098	+0.005
35	31.5	1.5	8	600	355	84	0.140	0.141	0.193	0.194	+0.001
35	31.5	0.5	10	600	272	78	0.104	0.117	0.152	0.165	+0.013
35	31.5	0.5	10	1000	438	77	0.163	0.178	0.205	0.221	+0.016
35	31.5	1.5	8	1000	542	84	0.189	0.201	0.236	0.247	+0.012
30	27.5	0.5	10	260	155	83	0.040	0.045	0.081	0.086	+0.005
30	27.5	1.5	8	202	159	83	0.048	0.047	0.089	0.088	−0.001
30	27.5	0.5	10	605	278	80	0.089	0.103	0.129	0.144	+0.014
30	27.5	1.5	8	595	378	72	0.130	0.144	0.161	0.175	+0.014
30	27.5	0.5	10	1007	438	91	0.135	0.148	0.179	0.191	+0.013
30	27.5	1.5	8	995	609	89	0.166	0.168	0.225	0.227	+0.002
25	22.3	0.5	10	258	155	86	0.034	0.034	0.069	0.069	0.000
25	22.3	1.5	8	200	160	86	0.039	0.037	0.074	0.072	−0.002
25	22.3	0.5	10	600	312	86	0.086	0.090	0.122	0.126	+0.004
25	22.3	1.5	8	592	382	83	0.106	0.110	0.140	0.144	+0.004
25	22.3	0.5	10	1007	462	85	0.120	0.127	0.154	0.161	+0.007
25	22.3	1.5	8	1007	607	84	0.151	0.153	0.184	0.185	+0.002
20	17.2	0.5	10	280	175	79	0.030	0.028	0.062	0.060	−0.002
20	17.2	1.5	8	250	192	81	0.044	0.041	0.069	0.066	−0.003
20	17.2	0.5	10	598	324	74	0.069	0.079	0.089	0.099	+0.010
20	17.2	1.5	8	604	410	74	0.090	0.094	0.111	0.115	+0.004
20	17.2	0.5	10	1015	496	79	0.094	0.107	0.115	0.128	+0.013
20	17.2	1.5	8	1000	645	74	0.122	0.127	0.140	0.147	+0.005
15	11.3	0.5	10	270	172	88	0.022	0.021	0.041	0.040	−0.001
15	11.3	1.5	8	200	166	85	0.022	0.020	0.040	0.038	−0.002
15	11.3	0.5	10	604	356	86	0.055	0.052	0.073	0.070	−0.003
15	11.3	1.5	8	599	431	84	0.066	0.062	0.082	0.078	−0.004
15	11.3	0.5	10	1000	560	82	0.066	0.074	0.082	0.091	+0.008
15	11.3	1.5	8	1000	682	85	0.083	0.082	0.101	0.100	−0.001

[a]Actual inside dimensions of flue lining 6½ × 11 in.

[b]Corrected for outside temperature 32 F. Barometer 29.92 in. Hg.

[c]½ gal per hour of fuel oil burned with 10 per cent CO_2 produces 18.4 cfm or 1.38 lb per minute of flue gases (corrected to 70 F, 1 atmosphere pressure); 1½ gal per hour of fuel oil burned with 8 per cent CO_2 produces 67.5 cfm or 5.06 lb per minute of flue gas (corrected to 70 F, 1 atmosphere pressure).

$$O_2 = O_1 + S_2 - S_1 \tag{12}$$

where

S_1 = computed static (theoretical) draft, experimental conditions.
S_2 = computed static (theoretical) draft, standard conditions.
O_1 = observed draft, experimental conditions.
O_2 = observed draft corrected to standard conditions.

It will be noted that the observed draft exceeded the computed static draft during some observations on the shorter chimneys. This is mainly attributed to the draft producing effect of the hot gases immediately above the chimney. By means of a manometer it was found that a measurable draft existed in this gas column for some distance above the chimney top. However, the temperatures in the chimney were measured with unshielded thermocouples and the actual gas temperatures may have been higher for this reason than the observed temperatures on which the computations of draft were based. The tests were made in calm weather.

Tests were made at the *National Bureau of Standards* to find the draft produced by round, metal smoke pipe set in vertical position to act as chimneys. Curves are presented in Fig. 5 showing the computed static or theoretical draft for short chimneys for various heights and temperatures. For this purpose, the density of chimney gases was assumed to be the same as that of air at the same condition, since the error thus introduced was not considered important in this case. The results of the tests showed that the following procedures would yield the available draft for 6-in. flue pipe used as a chimney within 10 per cent for the range shown and for fuel burning rates from about one-quarter to three-quarters of a gallon of oil per hour.

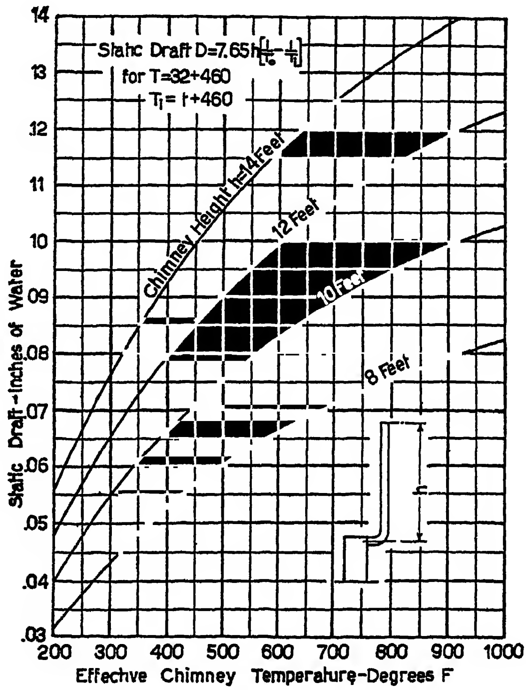


Fig. 5. Computed Static Draft for Short Chimneys

Using the temperature at the smoke collar of the heater, find the static draft corresponding to the available chimney height. Then:

1. If the chimney is bare, multiply the static draft by 0.76 to find the available draft.
2. If the chimney is insulated with 1 in. of air-cell material with a ¼-in. air space, multiply the static draft by 0.85 to find the available draft.
3. If the chimney is insulated with 1 in. of air-cell material with a 1-in. air space, open

at top and bottom for ventilation, multiply the static draft by 0.81 to find the available draft.

4. If the chimney is insulated with 1 in. of air-cell material and has a 1-in. air space closed at top and bottom to prevent ventilation, multiply the static draft by 0.85 to find the available draft.

The use of the 1-in. air-cell asbestos insulation in the tests discussed is not to be construed as an approval of such insulation in all cases in regard to fire resistance. Several laboratories are working on the fire resistance aspects of the problem but definite rules are not yet available. For coal- or oil-burning devices, a bare smoke pipe is probably safe if kept 2 ft or more from any woodwork and the better the pipe is insulated, or the lower its temperature, the nearer it can be placed to combustible materials.

DRAFT REQUIREMENTS OF DOMESTIC APPLIANCES

Typical flue-gas temperatures and drafts required at rated output for several kinds of domestic heating appliances[4] are contained in Table 3.

TABLE 3. DRAFTS REQUIRED BY TYPICAL DOMESTIC HEATING DEVICES OR APPLIANCES

DEVICE	DRAFT, INCHES WATER	STACK TEMPERATURE F DEG
Space Heater, Oil Burning, Pot Burner	0.06 to 0.08	1000
Warm Air Furnace, Oil Burning, Pot Burner	0.06	860
Warm Air Furnace, Hand Fired	0.06[b]	900
Floor Furnace, Oil Burning, Pot Burner	0.06	860
Mechanical Oil Burner, Less than 5 gph	0.03[a]	—
Mechanical Oil Burner, More than 5 gph	0.05[a] or less	—
Cooking Stove, Solid Fuel	0.04[b]	400
Space Heater, Coal Burning	0.06[b]	900

[a]Draft in fire-box. [b]For chestnut sized anthracite.

CHIMNEYS FOR GAS HEATING

Heating appliances designed to burn gas as well as appliances converted to gas burning, except those equipped with power type burners and excepting conversion burner installations in excess of 400,000 Btu per hour input in large steel boilers, are always equipped with a draft hood attached to the flue outlet of the appliance. This draft hood is required if the appliance is to meet the approval requirements of the *American Gas Association* and the *American Standards Association* and is essential for safe operation. It is designed to prevent excessive chimney draft which would lower appliance efficiency, to prevent a blocked flue or a down draft in the chimney from impairing combustion, to provide a relief opening for the products of combustion during down draft or blocked flue conditions, and to prevent spillage of the products of combustion to the space surrounding the appliance if there is a chimney draft equivalent to that provided by a 3-ft chimney. As the draft hood is designed without moving parts, the relief opening is always open and consequently some air is drawn into the chimney. While the air drawn in lowers the gas temperature in the chimney, it also lowers the dew-point of the gases and tends to prevent condensation.

The installation of conversion burner equipment in large boilers is usually made in accordance with regulations of the local gas company. In such installations a definite chimney draft may be required for proper

combustion and consequently the foregoing reference to the use of draft hoods would not apply.

The products of complete combustion of gas are water vapor (H_2O) and carbon dioxide (CO_2). In the case of manufactured gas, the presence of organic sulfur compounds, generally between 3 and 15 grains per hundred cubic feet, gives rise to minute percentages of sulfur dioxide and sulfur tri-oxide.

The volume of water vapor in the flue products from natural or coke oven gas is about twice the volume of carbon dioxide. It is extremely important that the chimney be tight and resistant to corrosion not only of moisture, but also of dilute sulfur trioxide.

Vitreous tile linings with joints which prevent retention of moisture and linings made of non-corrosive materials are advantageous. The protection of unlined chimneys has been investigated and the results indicate that after the loose material has been removed, the spraying with a water emulsion of asphalt chromate will provide excellent protection.

Advice regarding recommended practice and materials for flue connections and chimney linings can usually be obtained from the local gas company and should be given careful consideration.

Since a gas designed appliance must be able to operate at rated input (plus 10 or 15 per cent) without chimney connection, and without producing carbon monoxide, the only function of the chimney is to remove the products of combustion from the room. The chimney provides draft to overcome the friction in the flue pipe and chimney, but does not draw air into the appliance.

Chimneys for venting appliances designed for burning gas can therefore be low in height, but must have adequate area. The height is usually established by the building height. Chimney sizes are usually selected on the basis of Btu input of the appliance. One chart[5] designed to facilitate selection is shown in Fig. 6. The assumptions made in preparing the chart as well as its limitations should be noted carefully.

Since Fig. 6 has been prepared for circular flues, relative capacities for rectangular and semi-elliptical flues[6] are shown in Fig. 7.

When a flue is connected to several appliances, the number of horizontal runs of various sizes which may be substituted for the single run having a diameter equal to that of the flue may be obtained from Table 4.

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Building Code recommended by the *National Board of Fire Underwriters*, Article XI, Sections 1101 to 1105, in which the following are some of the important provisions listed in the 1943 edition:

(*a*) Chimneys erected within or attached to a structure shall be constructed of brick, of solid block masonry, or of reinforced concrete.

(*b*) Chimneys shall extend at least 3 ft above the highest point where they pass through the roof of the building and at least 2 ft higher than any ridge within 10 ft of such chimney.

(*c*) Every such chimney shall be properly capped with brick, terra cotta, stone, cast-iron, concrete or other approved non-combustible, weatherproof material.

(*d*) Chimneys shall be wholly supported on approved masonry or self-supporting fireproof construction.

(*e*) No such chimney shall be corbeled from a wall more than 6 in.; nor shall such chimney be corbeled from a wall which is less than 12 in. in thickness unless it projects equally on each side of the wall; provided that in the second story of two-story dwellings

corbeling of chimneys on the exterior of the enclosing walls may equal the wall thickness. In every case the corbeling shall not exceed 1 in. projection for each course of brick projected.

(*f*) No change in the size or shape of a chimney, where the chimney passes through the roof, shall be made within a distance of 6 in. above or below the roof joists or rafters.

(*g*) Smoke flues for warm air, hot water and low pressure steam heating furnaces shall have walls not less than 8 in. thick; the walls may be of solid masonry using brick, stone or concrete, or of solid moulded or solid cast chimney units of concrete, or of burned clay, or of suitably reinforced solid concrete cast in place; provided that for stone masonry, other than sawed or dressed stone in courses, the thickness shall be not less

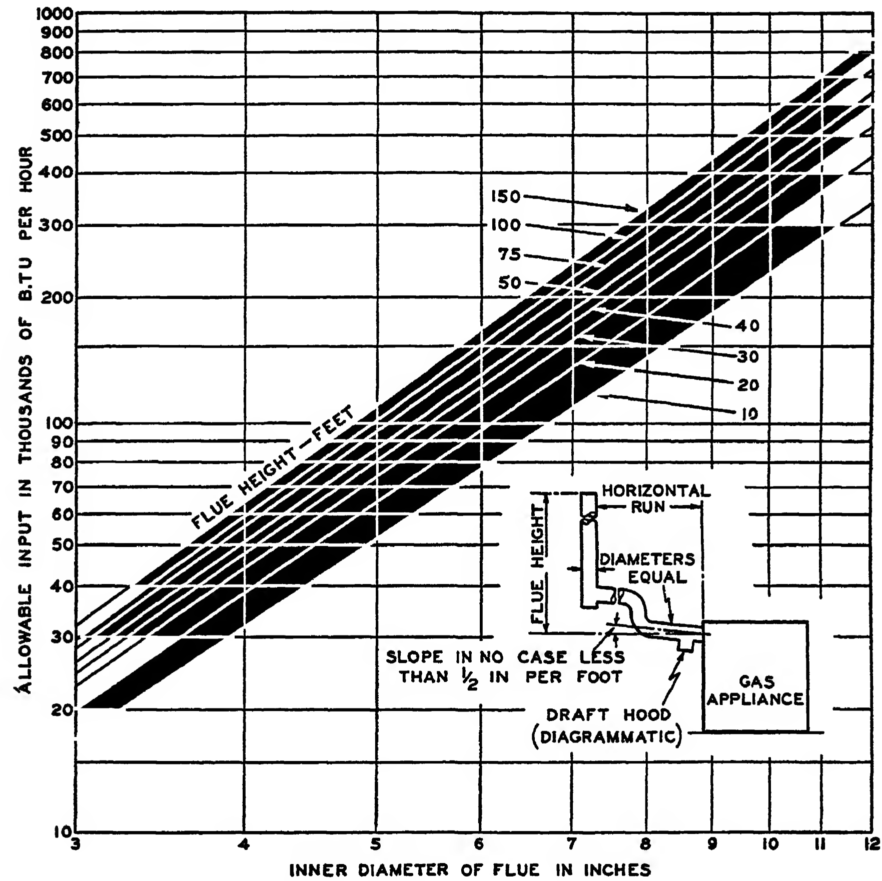


FIG. 6. ALLOWABLE BTU INPUT TO CIRCULAR FLUES FOR DOMESTIC GAS APPLIANCES WITH DRAFT HOODS

Notes Applying to Fig. 6:

1. Chart is based on: average flue temperature of 150 F, outside temperature of 60 F, barometric pressure of 30 in Hg, 100 per cent excess air and 100 per cent dilution at draft hood.

2. Based on terra-cotta lined flues. With rough brick flues, capacities are 15 per cent less.

3. Based on condition that horizontal run is not greater than 20 ft except for a flue height less than 20 ft, in which case the horizontal run is not to have greater length than the height of the flue.

4. Two long radius elbows are included in the horizontal run, the diameter of which is equal to that of the flue.

5. Each additional elbow reduces the allowable horizontal run by a length in feet equal to the diameter in inches.

6. When the horizontal run has an effective length in excess of that given (or additional elbows) the next larger size of flue should be chosen. It is desirable that long horizontal runs be insulated to reduce heat loss of flue products and to conserve draft.

7. Capacities should be reduced 3.5 per cent for each 1000 ft above sea level.

than 12 in. The walls shall be properly bonded, or tied with non-corrosive metal anchors. In dwellings and buildings of like heating requirements the thickness of the chimney walls may be reduced to not less than 3¾ in. when lined with a flue lining conforming to the Code requirements.

(*h*) Required flue linings shall be made of fire clay or other refractory clay to withstand the action of flue gases and to resist, without softening or cracking, the temperatures to which they will be subjected, but not less than 2,000 F, or of cast-iron of approved quality, form and construction. Approved corrosion resistant linings may be used in flues for gas appliances.

(*i*) Required clay flue linings shall be not less than ⅝ in. thick for the smaller flues and increasing in thickness for the larger flues.

(*j*) Flue linings shall be built ahead of the construction of the chimney as it is carried up, carefully bedded one on the other in mortar as hereinafter specified with close fitting joints left smooth on the inside.

(*k*) Flue linings shall start from a point not less than 8 in. below the intake. They

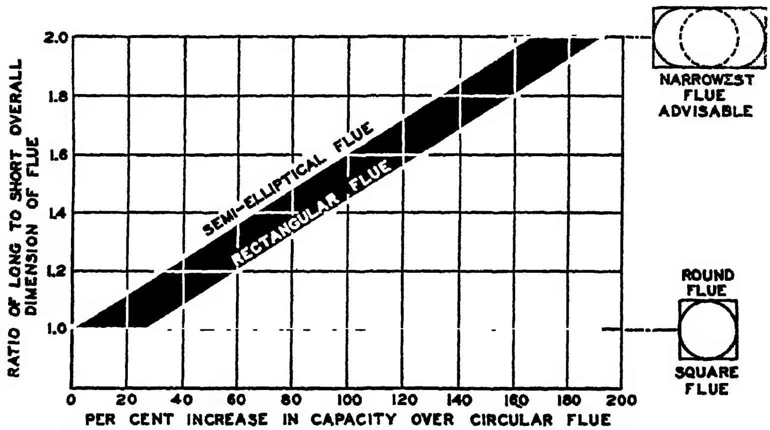


Fig. 7. Capacity of a Rectangular Flue or a Semi-Elliptical Flue, with Semi-Circular Ends Having Its Minimum Width Equal to the Diameter of a Circular Flue, Compared with the Capacity of the Circular Flue

shall extend, as nearly vertically as possible, for the entire height of the chimney. It is recommended that flue linings be extended 4 in. above the top or cap of the chimney.

(*l*) Only Portland cement mortar, cement lime mortar or fire clay mortar shall be used in setting flue linings.

For *gas appliances* the Building Code specifies lined chimneys and metal smoke stacks for all appliances which may be converted readily to the use of solid or liquid fuel and also for all boilers and furnaces except those having a flue gas temperature not exceeding 550 F at the outlet of the draft hood when burning gas at the manufacturer's rating and which may therefore be connected to *Type B* vent piping. Approved *Type B* vent piping is non-combustible, corrosion resistant piping of adequate strength and heat insulating value, and having bell and spigot or other acceptable joints. Fig. 6 may be used for selection of vent pipe size.

Important points to be considered in the use of *Type B* vent piping are:

1 *Type B* flues must be plainly and permanently marked, at the point where the vent connection enters the flue *For use of gas appliances only.*

2. *Type B* vent material should not be used for external chimney flues and external runs of it should not exceed 3 ft outside the building roof. When this requirement makes it necessary to cross over through attic space, the piping should be pitched not less than 45 deg.

3. Because of the small size and low temperature, *Type B* vents should be provided with a vent cap with wire screening to prevent building of birds' nests.

4. Each appliance should have the equivalent of a 4 in. diameter *Type B* vent, even though the appliance may have a 3 in. flue collar. A typical minimum vertical flue size for a frame dwelling is 6 in. in diameter or equivalent.

5. When several floor furnaces are to be vented, it is acceptable practice to connect each of these by means of 4 in. vents to a common 6 in. vertical vent. Lateral piping must have adequate pitch, ½ in per foot, and should not exceed 20 ft in horizontal length.

6. An alternate method of connecting several appliances is to run separate *Type B* vents to the attic and then to connect them by means of cross-over piping and Y fittings to a common vertical vent passing through the roof. This reduces the number of holes in the roof.

All flue mortar for flues or vent pipes from gas burning appliances shall be acid resisting.

GENERAL CONSIDERATIONS FOR CHIMNEYS

The draft of domestic chimneys may be subject to a variety of influences not usually encountered in power chimneys[7]. Horizontal winds have an aspirating effect as they cross the chimney and are an aid to draft. However, surrounding objects, such as trees or other buildings, may

TABLE 4. EQUIVALENT FLUE PIPE SIZES[a]

Diameter of Horizontal Runs	Size of Flue						
	3	4	5	6	8	10	12
3	1	2	3	5	9	12	22
4		1	2	3	5	7	11
5			1	2	3	4	7
6				1	2	3	5
8					1	2	3

[a]Comfort Heating (*American Gas Association*).

affect the direction of the wind at the chimney top and may even direct it down the chimney, tending to reduce the draft or even to cause it to be negative. Although the chimney should extend well above the highest part of the roof, it is impracticable to carry it much beyond this point.

It is also important to consider the source of the air supply for proper combustion. Usually the boiler or furnace is located in the basement. When the furnace room has windows or doors opening to the outside on two or more sides of the house, the leakage of air will be sufficient for combustion, even though the windows and doors may be shut. If, however, the leakage is not sufficient to prevent an appreciable drop of pressure in the furnace room below that of the air outside, the chimney draft will be reduced by the difference between the atmospheric pressure outside and that inside the boiler room. In case the boiler room is fairly tight and is open to the outside on only one side of the house, then the draft will be affected in windy weather even with windows or doors open. If the wind is blowing toward the boiler room the draft will be increased, but if blowing in the opposite direction the draft may be decreased.

It is not to be assumed that increasing the cross-section area of a chimney will always effect a cure for poor draft. The opposite result may be experienced because of the cooling effect of the larger area. This reduces the theoretical draft and the velocity of the gases, and affords a greater opportunity for counter currents in the chimney. Sometimes the only practical remedy for a chimney with bad draft, when the chimney is of the proper size and is affected by conditions beyond control, is to resort to mechanical draft. This can often be done at small expense and the arrangement can be such that the fan or blower need be operated only when conditions are bad.

Two or more chimneys, either large or small, should never be connected together. If connected at the bottom, hot gases in the inverted U-tube thus formed would be in unstable equilibrium. Cold air would descend through one such chimney, from the top, and drive the hot gases out of the other and thus annul the draft.

More than one device can be served by one chimney. Batteries of boilers are commonly connected to a single chimney in power plants. However, if two or more chimneys are used, each chimney should be used separately for part of the boilers, and not connected in manifold with another chimney, in order to avoid the difficulty described previously.

In domestic installations it is sometimes necessary to serve a space heater or cooking stove and a water heater with the same chimney flue. This is not desirable, especially for low chimneys, since doors left open on one device while it is un-fired will tend to annul the draft on another device. Gas burning devices, with their draft hoods and lack of draft dampers, are especially bad in this respect. The traditional method of avoiding this with brick chimneys has been to construct multiple-flue chimneys, so that each fuel-burning device could be served by a separate opening. If two devices must be served by one flue-opening in a chimney, their connections to the chimneys should not be located opposite each other. The connection from the larger device should be reasonably low down and that from the smaller, up near the ceiling, so that each device can be serviced as well as possible, regardless of the treatment of the other.

Excessive height in a chimney does no harm but means for controlling the draft are more than ordinarily essential if the chimney is too large in capacity. Coal-burning devices often have air leaks around the fire-box and the draft doors sometimes fit poorly so that the fire cannot be controlled at a low rate. Perhaps the simplest remedy for such cases is the barometric damper which admits air into the flue pipe and thus reduces the draft.

Directions for building chimneys for fireplaces are contained in Department of Agriculture Farmers' Bulletin No. 1889.

It is considered bad practice to connect any heating device to a fireplace flue unless the fireplace is effectively sealed.

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CHAPTER 20

Estimating Fuel Consumption For Space Heating

Fuel Consumption Records, Calculated Heat Loss Method, Degree-Day Method, Estimating Fuel Consumption, Degree-Day as an Operating Unit, Maximum Demands and Load Factors, Seasonal Efficiency

MANY methods are in use for estimating in advance of actual operation the anticipated heat or fuel consumption of heating plants over long or short periods. With suitable modification in procedure these same general methods are frequently useful in checking the degree of effectiveness with which heat or fuel is utilized during plant operation.

In applying any of these estimating methods to the consumption of a particular building plant it should be noted that (*a*) reliable records of past heat or fuel consumptions of the building under consideration will usually produce more trustworthy estimates of future consumptions than will any data obtained by averages or from other similar buildings; (*b*) where no past records exist useful data can sometimes be obtained from records of similar types of buildings with similar plants *in the same locality;* (*c*) records of consumption, which are averages from many types of plants in many types of buildings in various localities, can produce no better than an average estimate which may be far from accurate; (*d*) estimates based on computed heat losses without the benefit of operating data are wholly dependent on how well the computation represents the actual facts.

Estimates based on computed heat losses alone are especially necessary where unusual operating conditions are encountered such as excessive ventilation, abnormal inside temperatures, heat gains from external sources, etc., or where no information is available as to former consumption as in the case of proposed buildings of unusual design.

In interpreting and evaluating heat or fuel consumption estimates as well as in their preparation, it is well to realize that any estimating method used will produce a more reliable result over a long period operation than over a short period. Nearly all of the methods in common use will give trustworthy results over a full *annual* heating season, and in some cases such estimates will prove consistent within themselves for monthly periods. As the period of the estimate is shortened there is more chance that some factor not allowed for in the estimating method will become controlling and thus give discrepant and even ridiculous results.

Of the various estimating methods in use attention is directed in this discussion to but two as they are illustrative of all, viz: (1) calculated heat loss method, and (2) degree-day method.

CALCULATED HEAT LOSS METHOD

This method is theoretical and assumes constant temperatures for very definite hours each day throughout the entire heating season. It does not take into account factors which are difficult to evaluate such as opening of windows, abnormal heating of the building, poor heating systems, winter heat gains, such as sun effect, and many others. In order to apply this method the hourly heat loss from the building under maximum load,

or design condition, is computed following the principles discussed in Chapters 6 and 8 and the method described and illustrated in Chapter 14.

In predicting fuel consumption for heating a building by the Calculated Heat Loss Method, the general equation is:

$$F = \frac{H\,(t - t_a)\,N}{E\,(t_d - t_o)\,C} \tag{1}$$

where

F = quantity of fuel or energy required (in the units in which C is expressed).

H = calculated heat loss, Btu per hour, during the design hour, based on t_o and t_d $\left(\text{generally } H = H_t + H_i \text{ but may on occasion equal } H_t + \frac{H_i}{2}\right)$.

t = average inside temperature maintained during heating period, Fahrenheit degrees.

t_a = average outside temperature through estimate period, Fahrenheit degrees (for cities with an Oct. 1-May 1 heating season, see Table 1, Chapter 14).

t_d = inside design temperature, Fahrenheit degrees (usually 70 F).

t_o = outside design temperature, Fahrenheit degrees (see Table 1 in Chapter 14).

N = number of heating hours in estimate period (for an Oct. 1—May 1 heating season, 212 days × 24 hr = 5088).

E = efficiency of utilization of the fuel over the period, expressed as a decimal; not the efficiency at peak or rated load condition.

C = heating value of one unit of fuel or energy.

Although the assumption of an Oct. 1-May 1 heating season is reasonably accurate in the well-populated New York-Chicago zone, it is not valid as far north as Minneapolis nor farther south than Washington, D. C. and St. Louis. Consequently, it is suggested that allowance be made for this variation, especially in the far north or southern cities.

Example 1. A residence building is to be heated to 70 F from 6 A.M. to 10 P.M and 55 F from 10 P.M. to 6 A.M. The calculated hourly heat loss is 120,000 Btu per hour based on 70 F inside at −10 F outside. If the building is to be heated by metered steam, how many pounds would be required during an average heating season?

Solution. The heating value of steam may be taken as 1000 Btu per pound, and since it is purchased steam, the efficiency can be assumed as 100 per cent. Assume average outside temperature as 36.4 F. The average inside temperature is:

$$\frac{(16 \times 70) + (8 \times 55)}{24} = 65 \text{ F.}$$

Substituting in Equation 1:

$$F = \frac{120{,}000\ (65 - 36.4)\ 5088}{1.00\ [70 - (-10)]\ 1000} = 218{,}275 \text{ lb.}$$

Example 2. How much would the fuel cost to heat the building in Example 1 during an average heating season with coal at $8 per ton and with a calorific value of 11,000 Btu per pound, assuming that the seasonal efficiency of the plant was 55 per cent?

Solution. Substituting in Equation 1: $F = \dfrac{120{,}000\ (65 - 36.4)\ 5088}{0.55\ [70 - (-10)]\ 11{,}000} = 36{,}079$ lb = 18 tons, which, at $8 per ton, costs $144.

Example 3. What will be the estimated fuel cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outside temperature during the heating season is 36.4 F. The seasonal efficiency will be 75 per cent. The heating plant will be thermostatically controlled, and a temperature of 55 F will be maintained from 11 P.M. to 7 A.M. Assume that the price of gas is 7 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

Solution. The average hourly temperature is:

$$t_a = \frac{(72 \times 16) + (55 \times 8)}{24} = 66.3 \text{ F.}$$

The maximum hourly heat loss will be:

$$H = 92{,}000 - \frac{26{,}000}{2} = 79{,}000 \text{ Btu.}$$

$$M = \frac{79{,}000\ (66.3 - 36.4) \times 24 \times 210}{100{,}000 \times 0.75 \times (72 - 0)} = 2204.6 \text{ hundred thousand Btu.}$$

$2204.6 \times \$0.07 = \$154.32$ = estimated fuel cost per year of heating building.

Several time-saving procedures have been devised for quickly estimating the hourly Btu loss of one and two-story residences in order that fuel estimates can be predicted more quickly from Equation 1. A graphical method of calculating heat losses has been developed [1] which makes possible a quick solution if the gross wall, ceiling, or floor areas and respective transmission coefficients are known.

The Federal Housing Administration has originated a short-cut formula for residential heat loss determinations which makes use of the floor area and three selected transmission coefficients. The formula was developed to apply to detached houses approximately rectangular in shape with total exterior door and window areas equal to about 25 per cent of the floor area and with a floor area not greater than about 1500 sq ft. Equation 2 is for a one-story residence and Equation 3 is intended for two-story structures.

$$H_1 = A\ (G + U_w + U_c + U_f)\ (t_d - t_o) \tag{2}$$

$$H_2 = A\ (G + 1.2\ U_w + 0.5\ U_c + 0.5\ U_f)\ (t_d - t_o) \tag{3}$$

where

H_1 = heat loss from one-story residence, Btu per hour.
H_2 = heat loss from two-story residence, Btu per hour.
A = floor area, square feet, measured to the inside faces of enclosing walls and is the sum of the following areas: (1) all the area on each principal floor level; (2) the area of all finished habitable attic rooms, including bathrooms, toilet compartments, closets, and halls; (3) all other areas intended to be heated and not located in the basement.
G = glass and infiltration factor for ordinary construction: (0.45 for no weatherstripping or storm windows), (0.40 for weatherstripping), (0.30 for storm windows with or without weatherstripping).
U_w = coefficient of transmission for outside wall.
U_c = coefficient transmission for ceiling.
U_f = coefficient of transmission for floor.
t_d = inside design temperature, Fahrenheit degrees.
t_o = outside design temperature, Fahrenheit degrees.

Notes for application of Equations 2 and 3.

1. The calculation of heat loss from heated spaces into adjacent spaces such as attics, basementless areas, and heated or unheated garages shall be based on the assumption that the temperature of such adjacent spaces is the same as the outside design temperature.
2. For all floors over basements or other warmed spaces assume $U_f = 0$.
3. For structures having concrete slab floors laid on the ground a modified application of the formula may be made. Assume $U_f = 0$ and calculate the heat loss in accordance with the check formula. Then add the slab loss determined in accordance with the procedure developed by the National Bureau of Standards and described in BMS Report 103.
4. No basement area is to be included in the formula calculation. If finished habitable rooms in the basement are to be heated, the additional heat loss should be calculated separately and added to the amount obtained by the formula.

Both the graphical method and short-cut formulas, when used within the limitations established, have been found to give reasonably accurate

results for the average residence, but if precise estimates are required, the procedure outlined in Chapter 14 should be used.

In the case of gravity warm air heating installations, the load was formerly expressed in square inches of leader pipe which can be converted into Btu per hour by multiplying the square inches of leader area by 111, 167, and 200 for first, second, and third floor respectively.

Example 4. What would be the total gas consumption over a full heating season of a gas-fired gravity warm air furnace designed according to the Code[2], and with four 12 in. and two 8 in. round leaders to the first floor and six 10 in. leaders to the second floor, if the gas has a heating value of 500 Btu per cubic foot, the plant operates at a 70 per cent seasonal efficiency and is designed to maintain an average inside temperature of 65 F when it is 10 F outside in a city where the average outside temperature is 45 F and the heating season is 5088 hr long?

Solution. The area of the round leaders is: 12 in., 113 sq in.; 10 in., 79 sq in.; and 8 in., 50 sq in. The total Btu transmitted is:

First Floor: $[(4 \times 113) + (2 \times 50)] \times 111 = 61{,}272$ Btu per hour.
Second Floor: $(6 \times 79) \times 167 = 79{,}158$ Btu per hour.

Total 140,430 Btu per hour.

Substituting this total heat loss value as H in Equation 1 gives:

$$F = \frac{140{,}430\ (65 - 45)\ 5088}{0.70\ (70 - 10)\ 500} = 680{,}483 \text{ cu ft gas.}$$

DEGREE-DAY METHOD

This method is based on consumption data which have been taken from buildings in operation, and the results computed on a degree-day basis. While this method may not be as theoretically correct as the Calculated Heat Loss Method, it is considered by many to be of more value for practical use.

The amount of heat required by a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically it is proportional to the difference between the outdoor and indoor temperatures. The *American Gas Association* [3] determined from experiment in the heating of residences that the gas consumption varied directly as the difference between 65 F and the mean outside temperature. In other words, on a day when the mean temperature was 20 deg below 65 F, twice as much gas was consumed as on a day when the temperature was 10 deg below 65 F. For any one day, when the mean temperature is less than 65 F, there are as many degree-days as there are degrees difference in temperature between the mean temperature for the day and 65 F. Degree-days may be calculated on other than the 65 F base but are seldom used and are of little value except where the inside temperature to be maintained as, for example, in warehouses, differs greatly from the usual inside temperature range of 68 F to 72 F.

The normal or average number of degree-days, on a 65 F basis, which have occurred over a long period of years, by months, on a 65 F basis are given for various United States and Canadian and Newfoundland cities in Table 1. The United States values were computed from daily mean temperatures recorded by the Weather Bureau over a 43-year period [4] from 1899 to 1941. The number of degree-days for a calendar day of a given year was obtained by taking the difference between 65 F and the mean temperature determined from a reading of the maximum and minimum temperature for a particular locality. The daily normal was established by taking an average of the 43 daily degree-day figures. These

daily values were then added to obtain a monthly normal, and the yearly or seasonal degree-day figure was established by taking a summation of the monthly values. In general, attempts to apply the degree-day method to fuel consumption over a period of less than a month are of questionable value.

If the degree-days occurring each day are totaled for a reasonably long period, the fuel consumption during that period as compared with another period will be in direct proportion to the number of degree-days in the two periods. Consequently, for a given installation, the fuel consumption can be calculated in terms of fuel used per degree-day for any sufficiently long period and compared with similar ratios for other periods to determine the relative operating efficiencies with the outside temperature variable eliminated.

Studies made by the *National District Heating Association*[5] of the metered steam consumption of 163 buildings located in 22 different cities and served with steam from a district heating company substantiate the fact that the 65 F base originally chosen by the gas industry is approximately correct.

Formula for Degree-Day Method

The general equation for calculating the probable fuel consumption by the degree-day method is:

$$F = U \times N \times D \qquad (4)$$

where

F = fuel consumption for the estimate period.
U = unit fuel consumption, or quantity of fuel used per (degree-day) (*building load unit*).
N = number of *building load units* (when available use calculated hourly heat loss instead of actual amount of radiation installed).
D = number of degree-days for the estimate period.

Values of N depend on the particular building for which the estimate is being prepared and must be found by surveying plans, by observation, or by measurement of the building. Values of U for use in this equation are the unit fuel consumptions per degree-day and are obtained as a result of the collection of operating information. Certain of this information is presented later but before referring to these data attention is directed to the nature of the unit.

Unit Fuel Consumptions per Degree-Day

The quantity of fuel used per degree-day in a given heating plant can be reduced to a unit basis in terms of quantity of fuel or steam per degree-day per square foot of radiation, per cubic foot of heated building space, or per thousand Btu hourly heat loss at design conditions. A less frequently used basis is quantity of fuel per (degree-day) (square foot of floor area). In fact any convenient unit can be used to relate the consumption to the degree-day and to the building.

The choice of these units requires explanation and some discrimination and judgment. If the volume basis is used, the net heated space is preferable to the gross building cubage since gross cubage includes outer walls and certain portions of attic and basement space which are usually unheated. In the absence of data on net heated volume a figure of 80 per cent of the gross volume may be used to obtain the estimated net heated volume. The volume basis has been rather widely used primarily because

Table 1. Normal Degree-Days for Cities in the United States, Canada and Newfoundland[a]

State	City	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug	Sept.	Oct.	Nov.	Dec.	Total
Ala.	Birmingham	586	502	318	133	23	1	0	1	10	111	351	582	2618
	Mobile	392	317	178	54	3	0	0	0	1	45	203	374	1567
Ariz.	Phoenix	404	261	152	47	7	0	0	0	0	18	168	389	1446
Ark.	Fort Smith	761	623	393	159	34	1	0	0	12	130	409	708	3230
	Little Rock	700	584	370	148	30	1	0	0	10	121	382	659	3005
Calif.	Los Angeles	276	232	209	158	104	27	1	0	5	43	110	225	1390
	San Francisco	464	340	314	269	252	196	198	180	122	140	242	426	3143
Colo.	Denver	1026	900	792	523	271	62	9	8	124	415	721	1012	5863
	Grand Junction	1230	887	667	381	152	22	1	1	60	352	745	1149	5647
Conn.	New Haven	1103	1024	848	521	222	46	3	12	89	341	660	1010	5879
D.C.	Washington	929	842	637	345	104	14	0	2	43	254	558	870	4598
Fla.	Jacksonville	295	241	131	39	3	0	0	0	0	24	142	286	1161
Ga.	Atlanta	662	568	389	173	32	2	0	1	12	130	393	640	3002
	Savannah	395	335	200	68	6	1	0	0	1	46	209	386	1647
Idaho	Boise	1065	840	680	441	248	87	10	17	135	394	717	1025	5659
Ill.	Chicago	1219	1084	871	536	260	67	7	8	84	334	709	1108	6287
	Springfield	1145	988	732	382	130	15	1	3	64	290	666	1047	5463
Ind.	Evansville	942	826	588	289	82	5	0	1	32	207	540	875	4387
	Indianapolis	1104	982	751	414	155	22	2	4	65	299	663	1026	5487
Iowa	Des Moines	1326	1143	849	449	165	27	2	6	99	357	768	1200	6391
	Sioux City	1410	1216	913	488	194	37	3	10	124	405	841	1268	6909
Kan.	Dodge City	1050	878	669	354	134	16	1	3	58	276	644	994	5077
	Topeka	1104	932	666	331	112	11	0	2	54	258	623	1008	5101
Ky.	Lexington	964	862	650	353	123	14	1	3	47	258	600	916	4791
	Louisville	931	824	595	302	92	7	0	1	34	218	549	875	4428
La.	New Orleans	319	251	130	32	1	0	0	0	0	23	147	305	1208
	Shreveport	522	421	243	83	10	0	0	0	4	71	274	499	2127
Me.	Eastport	1353	1242	1090	778	532	297	159	148	274	528	831	1219	8451
	Portland	1296	1184	1003	668	375	135	29	46	179	459	792	1172	7338
Md.	Baltimore	922	844	650	347	98	12	0	2	34	230	531	852	4522
Mass.	Boston	1101	1027	852	538	248	66	8	16	98	338	651	1000	5943
Mich.	Detroit[b]	1228	1141	943	569	252	56	7	15	109	387	753	1120	6580
	Marquette	1471	1358	1208	801	493	220	87	102	257	556	927	1306	8786
Minn.	Duluth	1726	1504	1267	809	514	232	82	100	289	636	1066	1541	9766
	Minneapolis	1597	1379	1082	580	255	59	8	24	163	484	943	1415	7989
Miss.	Vicksburg	498	413	239	85	9	0	0	0	5	77	268	479	2073
Mo.	Kansas City	1083	921	657	327	106	11	0	2	50	242	599	986	4984
	St. Louis	999	865	616	305	88	7	0	1	36	218	558	917	4610
	Springfield	971	835	599	306	105	11	1	2	45	232	560	900	4567
Mont.	Havre	1548	1374	1109	623	338	129	28	56	275	604	1010	1380	8474
Neb.	Lincoln	1254	1070	798	411	161	23	1	6	84	328	732	1142	6010
	Omaha	1285	1098	813	414	152	21	1	4	84	326	741	1163	6102
Nev.	Winnemucca	1126	887	763	539	320	109	11	23	190	502	802	1099	6371
N. H.	Concord	1339	1212	1005	633	310	98	19	46	189	492	827	1221	7391
N. J.	Atlantic City	944	893	762	489	215	37	1	2	40	250	552	864	5049
N. M.	Santa Fe	1094	892	786	544	297	62	10	15	129	451	772	1072	6124
N. Y.	Albany	1276	1174	955	549	221	44	4	14	116	410	756	1139	6658
	Buffalo[b]	1220	1168	1010	672	352	90	15	24	125	410	747	1102	6935
	New York	1025	958	785	467	176	29	1	4	51	276	600	934	5306
N. C.	Raleigh	699	618	436	213	47	5	0	1	17	155	417	673	3281
	Wilmington	528	481	328	147	25	2	0	0	5	92	310	514	2432
N. D.	Bismarck	1720	1478	1187	650	335	104	20	44	240	605	1059	1527	8969
Ohio	Cincinnati	1009	905	681	374	132	16	1	3	52	274	613	953	5013
	Cleveland	1141	1074	891	558	255	56	8	14	92	354	688	1040	6171
	Columbus	1082	982	762	436	167	26	2	6	68	318	674	1013	5536
Okla.	Oklahoma City	850	696	459	204	56	3	0	0	22	156	461	791	3698
Ore.	Baker	1221	994	839	600	411	210	56	73	261	540	849	1165	7219
	Portland	776	622	530	370	241	106	29	29	109	298	540	729	4379
Pa.	Philadelphia	957	885	696	379	117	16	0	2	36	236	548	877	4749
	Pittsburgh[b]	1043	971	767	448	170	29	3	7	69	324	656	979	5466
S. C.	Charleston	450	386	245	84	7	1	0	0	1	48	227	421	1870
	Columbia	568	488	312	129	18	1	0	0	6	98	330	554	2504

TABLE 1. NORMAL DEGREE-DAYS FOR CITIES IN THE UNITED STATES, CANADA AND NEWFOUNDLAND[a] (CONCLUDED)

State Province	City	Jan.	Feb	Mar.	Apr.	May	June	July	Aug.	Sept.	Oct.	Nov	Dec.	Total
S. D.	Huron	1586	1366	1044	579	264	66	9	21	174	507	966	1415	7997
	Rapid City	1288	1152	980	607	333	104	15	29	188	503	843	1183	7225
Tenn.	Knoxville	774	669	478	229	55	3	0	0	20	189	497	751	3665
	Memphis	709	605	389	159	32	1	0	0	13	127	383	660	3078
	Nashville	785	681	476	222	54	2	0	0	19	170	470	741	3620
Tex.	El Paso	611	433	288	107	15	1	0	0	6	90	369	618	2538
	Fort Worth	582	470	269	100	16	1	0	0	5	76	288	549	2356
	Houston	366	279	144	31	3	0	0	0	1	29	164	343	1360
	San Antonio	382	290	148	38	4	0	0	0	0	32	169	361	1424
Utah	Modena	1191	944	811	569	336	93	7	11	156	503	833	1151	6605
	Salt Lake City	1084	868	708	450	232	62	4	5	95	377	712	1040	5637
Vt.	Burlington	1444	1327	1117	679	330	98	22	47	196	510	866	1294	7930
Va.	Lynchburg	829	735	545	290	82	12	1	2	37	232	524	793	4082
	Norfolk	709	654	492	257	65	6	0	0	9	132	397	664	3385
	Richmond	812	727	548	281	74	9	0	1	27	199	491	775	3944
Wash.	Seattle	765	644	595	440	304	162	71	71	174	372	558	708	4864
	Spokane	1136	935	748	488	282	113	21	37	185	483	816	1061	6305
W. Va.	Elkins	1030	950	774	492	233	59	15	23	114	403	724	997	5814
	Parkersburg	975	888	671	371	129	170	1	3	55	287	619	922	5091
Wis.	Green Bay	1498	1337	1099	663	325	89	17	39	176	495	891	1327	7956
	LaCrosse	1500	1291	1001	533	222	50	7	22	154	459	865	1338	7442
	Milwaukee	1329	1181	969	621	340	100	13	18	120	409	788	1198	7086
Wyo.	Cheyenne	1188	1070	994	726	455	160	42	48	247	592	880	1147	7549
	Lander	1423	1198	1000	674	406	150	28	44	263	630	1020	1401	8237
Alta.	Calgary	1674	1428	1240	750	496	270	124	186	450	744	1170	1395	9,927
	Edmonton	1829	1512	1302	720	434	270	124	186	450	713	1230	1519	10,289
B. C.	Vancouver	899	756	713	510	341	180	62	31	270	496	660	837	5,755
Man.	Winnipeg	2139	1820	1581	810	465	90	.....	62	270	744	1320	1829	11,130
N.B.	Moncton	1519	1428	1178	810	465	210	.....	93	300	620	930	1333	8,886
N. S.	Halifax	1302	1176	1085	780	496	210	.....	.....	210	496	780	1147	7,682
Ont.	Ottawa	1674	1484	1271	690	279	30	.....	.....	210	589	990	1457	8,674
	Port Arthur	1829	1624	1426	900	558	240	62	86	360	713	1140	1550	10,488
	Toronto	1333	1204	1209	720	372	60	.....	.....	180	558	870	1209	7,715
P.E.I.	Charlottetown	1178	1120	1209	870	529	210	.....	.....	600	558	870	1240	8,384
Que.	Montreal	1581	1428	1209	720	310	.....	.....	.....	180	558	960	1395	8,341
	Quebec	1705	1484	1333	870	434	120	.....	31	270	651	1050	1519	9,467
Sask.	Saskatoon	2108	1820	1581	810	465	210	62	155	450	806	1290	1736	11,493
Newf.	St.John's	1286	1215	1153	900	680	412	202	175	339	613	834	1110	8,919

[a]Computed from daily mean temperatures recorded by U. S. Weather Bureau over a 43-year period from 1899 to 1941. Data for Canadian cities abstracted from *Heating & Ventilating*, October, 1939. The National Joint Committee on Weather Statistics, in cooperation with the U. S. Weather Bureau, is preparing revised Degree-Day Normals.

[b]Data for these cities and possibly other localities are based on readings taken at more than one official Weather Station during the 43-year period of analysis and are subject to local verification, as these figures are being examined for possible revision by the U. S Weather Bureau.

of its facility in application. In industrial buildings it is usually easier to obtain the correct volume of a given building than to measure and evaluate the heating capacity of its heating system or calculate its maximum hourly Btu loss. The comparison of buildings on a straight volume basis does not allow for variation in exposure, type of construction, ratio of exposed area to cubical contents, and type of occupancy. It is considered inaccurate for purposes of estimating fuel consumption unless the buildings are of very similar nature.

The calculated heat loss or its equivalent square feet of calculated radiator surface may be used as the unit. The use of the unit equivalent direct radiation is of questionable value when referring to heat transfer

TABLE 2. UNIT FUEL CONSUMPTION CONSTANTS (U) FOR GAS[a]

Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 60 F.

Heating Value of Gas Btu per Cu Ft	Hot Water: Cu Ft Gas per Degree-Day per Sq Ft Radiator			Steam: Cu Ft Gas per Degree-Day per Sq Ft Radiator			Warm Air: Cu Ft Gas per Degree-Day per 1000 Btu Hourly Design Heat Loss	
	Up to 500 Sq Ft	500 to 1200 Sq Ft	Over 1200 Sq Ft	Up to 300 Sq Ft	300 to 700 Sq Ft	Over 700 Sq Ft	Gravity	Fan Systems
500	0.142	0.135	0.128	0.242	0.231	0.220	0 855	0.820
535	0.132	0.126	0.120	0.226	0.215	0.206	0.800	0.766
800	0.089	0.085	0.081	0.151	0.144	0 137	0 534	0.513
1000	0.071	0.068	0.065	0.121	0.115	0.110	0 428	0.410
1 Therm = 100,000 Btu	Gas Consumption in Therms per Degree-Day							
	0.000708	0.000675	0.000642	0.00121	0.00115	0.00110	0.00428	0.00409

[a]Abstracted from Comfort Heating, *American Gas Association*, 1938.

TABLE 3. UNIT FUEL CONSUMPTION[a] CONSTANTS (U) FOR OIL[b]

Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 55 F

Unit[c]	Efficiency in Per Cent				
	40	50	60	70	80
Gal Oil per Sq Ft Steam Radiator..	0.00172	0.00137	0.00114	0.00098	0.00086
Gal Oil per Sq Ft Hot Water Radiator	0.00108	0.00086	0.00072	0.00062	0.00054
Gal Oil per 1000 Btu per Hour Heat Loss	0.00715	0.00571	0.00476	0.00409	0.00358

[a]Based on a heating value of 140,000 Btu per gallon.

[b]Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss.

[c]Per degree-day.

TABLE 4. UNIT FUEL CONSUMPTION[a] CONSTANTS (U) FOR COAL[b]

Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 55 F.

Unit[c]	Efficiency in Per Cent				
	40	50	60	70	80
Lb Coal per Sq Ft Steam Radiator..	0.0200	0.0160	0.0133	0.0114	0.0100
Lb Coal per Sq Ft Hot Water Radiator	0.0125	0.0100	0.0084	0.0072	0.0063
Lb Coal per 1000 Btu per Hour Heat Loss	0.0825	0.0666	0.0550	0.0471	0.0412

[a]Based on a heating value of 12,000 Btu per pound

[b]Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss.

[c]Per degree-day.

surfaces used in warm air furnace or central air conditioning systems. Where steam or hot water radiation is already installed, care should be exercised in using the unit equivalent direct radiation basis for estimating, since actual installed radiation may differ considerably from the exact radiation requirements. In view of all these considerations it is believed that the unit based on *thousands of Btu of hourly calculated heat loss for the design hour* is probably the most desirable, although the one most widely used seems to be *units of fuel per degree-day per square foot of equivalent direct radiator surface.* The equivalent heating load for the hot water supply is not included in the latter unit, but it generally includes the piping load.

Since this unit is the one most widely used at present the unit fuel consumptions given in succeeding paragraphs of this chapter make use of this unit to a considerable extent, although it should be understood that most of these units of consumption can be transposed as desired.

Estimating Gas Consumption

Values of the Unit Fuel Consumption Constant (U) for gas are given in Table 2 for various gas heating values, and different types and sizes of heating plants. They are based on an inside design temperature of 70 F and an outside design temperature of 0 F and apply only to these conditions. For other outside design conditions corrections must be made as given in Table 5.

The factors in Table 2, as corrected if necessary, are satisfactory for regions having 3500 to 6500 degree-days per heating season. In regions with less than 3500 degree-days the unit gas consumption is higher than given; where over 6500, the unit is less than given. Ten per cent addition or deduction in these cases is recommended by *A.G.A.* publications. Estimates for industrial buildings where low inside temperatures are maintained cannot be made from this table.

For gas heating values other than those given in Table 2, simply interpolate or extrapolate. It will also be noted that Table 2 applies only to small installations. In general the larger the installation the smaller the unit gas consumption becomes and the values in the table should be used with care, if at all, in large gas-burning installations.

Example 5. Estimate the gas required to heat a building located in Chicago, Ill., which has 6287 degree-days and a gas heating value of 800 Btu per cubic foot. The calculated heating surface requirements are 1000 sq ft of hot water radiation based on design temperature of −10 F and 70 F.

Solution. From Table 2, the fuel consumption for a design temperature of 0 F with 800 Btu gas is found to be 0.085 cu ft of gas per (degree-day) (square foot of hot water radiation). From Table 5, the correction factor is 0.875 for −10 F outside design temperature, hence $0.875 \times 0.085 = 0.07438$. By Equation 4,

$$F = 0.07438 \times 1000 \times 6287 = 467{,}000 \text{ cu ft.}$$

Estimating Oil Consumption

Unit fuel consumption factors for oil, similar to those for gas in Table 2, are given in Table 3. The factors in Table 3 apply only to an inside design temperature of 70 F and an outside design temperature of 0 F. For other outside design temperatures, the constants in Table 3 must be multiplied by the values in Table 5 as explained under Estimating Gas Consumption.

Values given in Table 3 assume the use of oil with a heating value of 140,000 Btu per gallon. For other heating values, multiply the values in

TABLE 5. CORRECTION FACTORS FOR OUTSIDE DESIGN TEMPERATURES[a]

OUTSIDE DESIGN TEMP. F DEG.	−20	−10	0	+10	20
Correction Factor	0.778	0.875	1.000	1.167	1.400

[a]The multipliers in Table 5, which are high for mild climates and low for cold regions are not in error as might appear. The unit figures in Tables 2, 3, and 4 are per square foot of radiator or thousand Btu heat loss per degree-day. For equivalent buildings and heating seasons, those in warm climates have lower design heat losses and smaller radiator quantities than those in cold cities. Consequently, the *unit* figure in quantity of fuel per (square foot of radiator) (degree-day), is larger for warm localities than for colder regions Since the northern cities have more radiator surface per given building and a higher seasonal degree-day total than cities in the south, the total fuel per season will be larger for the northern city.

Table 3 by the ratio of 140,000 divided by the heating value per gallon of fuel being used.

Example 6. Estimate the seasonal oil consumption of an oil-fired boiler in a building located in Minneapolis having a calculated heat loss of 192,000 Btu per hour, burning 144,000 Btu per gallon oil and operating at a seasonal efficiency of 60 per cent. The outside design temperature for Minneapolis is −20 F, and the inside design temperature is 70 F.

Solution. From Table 3, under 60 per cent efficiency and opposite the bottom column, the value of U is found to be 0.00476 gal per 1000 Btu hourly heat loss for 0 F outside temperature.

The correction factor for −20 F outside design temperature from Table 5 is 0.778. Solving, $0.778 \times 0.00476 = 0.00370$. Making a further correction for the heating value:

$0.0037 \times \frac{140,000}{144,000} = 0.0036$ gal per 1000 Btu per hour calculated heat loss per degree-day.

From Table 1, the normal degree-days for Minneapolis are 7989. Since U is expressed in 1000 Btu, N is equal to 192. Substituting in Equation 4:

$$F = 0.0036 \times 7989 \times 192 = 5525 \text{ gal.}$$

Estimating Coal or Coke Consumption

Coal or coke consumption estimates are made in exactly the same procedure as for oil. Values of U are given in Table 4 which only apply to inside design temperatures of 70 F and an outside design temperature of 0 F. A correction must be made for other conditions by use of the multiplying factors in Table 5. Data in Table 4 are based on 12,000 Btu per pound coal and for other heating values of coal they must be multiplied by the ratio of 12,000 divided by the heating value of fuel used.

Example 7. A building in Marquette, Mich., has an hourly heat loss at design conditions of 240,000 Btu per hour. Based on an inside design temperature of 70 F and an outside design temperature of −20 F, what will be the estimated normal seasonal coal consumption for heating if 12,000 Btu per pound fuel is burned at a 50 per cent seasonal efficiency, and what part of the total will be used during November, December, and January?

Solution. From Table 4, U is 0.0666 lb of coal per 1000 Btu per hour heat loss. Correcting for the outside design temperature of −20 F from Table 5, the value of U is $0.778 \times 0.0666 = 0.0518$. From Table 1, D is 8786 and from the problem, N is 240.

Substituting in Equation 4:

$$F = 0.0518 \times 240 \times 8786 = 109,200 \text{ lb.}$$

Fuel used over any period is, according to the theory of the degree-day, proportional to the number of degree-days during the period. From Table 1, the average numbers of degree-days for November, December, and January in Marquette are 927, 1306, and 1471, a total of 3704. The yearly total is 8786, so that during these three months the estimated consumption is:

$$\frac{3704}{8786} \times 109,200 = 46,200 \text{ lb.}$$

Estimating Steam Consumption

In estimating steam consumption the efficiency is generally assumed at 100 per cent. If for low pressure steam an average heating value of 1000 Btu per pound of steam is used no correction is necessary. In comparing values from different cities, correction should be made for design temperature (see Table 5) when the unit figures are in terms of square feet of radiation but not when the values are in terms of building volume or floor space.

Where the heat loss is calculated in Btu per (hour) (*degree difference in temperature*) the simple Equation 5 may be used:

$$F = \frac{H \times 24 \times D}{1000} \qquad (5)$$

where

F = pounds of steam required for estimate period.
H = calculated heat loss, Btu per (hour) (degree difference).
D = number of degree days for the period of estimation.
1000 = Btu delivered per pound of steam condensed.

In this method the number of degree-days automatically takes care of average inside and outside temperature difference. When degree days are taken from Table 1, an average inside temperature of approximately 65 F is assumed throughout the period. If an average inside temperature other than approximately 65 F is to be used, the number of degree-days should be obtained for the new base.

Example 8. An eight story building in Pittsburgh maintains daytime temperatures of 70 F but allows night temperature to drop to not lower than 60 F. Its calculated heat loss is 10,500 Btu per (hour) (*degree temperature difference*). What is the estimated average yearly steam consumption for building heating?

Solution. Since the average inside temperature is approximately 65 F, the degree-days from Table 1, based on 65 F may be used. Therefore, from Table 1, Pittsburgh has 5,466 degree-days per normal season. Inserting in Equation 5

$$F = \frac{10{,}500 \times 24 \times 5466}{1{,}000} = 1{,}377{,}432 \text{ lb of steam.}$$

Consideration has been given to the difference in steam utilization of different types of buildings and Table 6 shows actual average units for these various types. These figures were obtained from operating results in 896 buildings located in all sections of the United States. Being averages, and for small groups in each type, the figures may need considerable modification to allow for local variations. It should be especially noted that the steam used for heating hot water is not included in the values given in Table 6.

Example 9. A store in Philadelphia with a heating system designed to maintain 70 F inside in 0 F weather has 250,000 cu ft of heated space. What would be the estimated average yearly steam consumption of purchased steam for heating?

Solution. According to Table 6, a store would use 0.624 lb of steam per degree-day per 1000 cu ft heated space. From Table 1, Philadelphia has 4749 degree-days per normal year. Inserting in Equation 4:

$$F = 0.624 \times 250 \times 4749 = 740{,}000 \text{ lb of steam.}$$

Degree-Day as an Operating Unit

The degree day is also widely used as a means of comparing the efficiency of the fuel consumption of one period with another *for the same*

TABLE 6. STEAM CONSUMPTION OF BUILDINGS WITH VARIOUS TYPES OF OCCUPANCY[a]

Type of Building	No. Bldgs.	Average Volume Heated Space 1000 Cu Ft	Steam for Heating Lb per DD per 1000 Cu Ft	Average Hours of Occupancy
Office	334	2160	0.685	12.1
Office and Bank	49	3000	0.577	13.1
Office and Printing	8	1895	1.230	17.7
Office and Theater	7	4950	0.412	12.9
Office and Stores or Shops	26	1615	0.617	13.2
Bank	16	806	0.786	11.7
Department Store	63	3400	0.385	11.1
Stores	73	310	0.624	10.4
Loft	63	865	0.588	10.0
Warehouse	24	2230	0.459	9.4
Hotel and Club	73	1795	0.990	22.3
Apartment or Residence	51	1425	0.962	21.8
Theater	22	1240	0.482	12.9
Garage	13	1540	0.202	21.4
Manufacturing	19	1350	0.808	9 5
Church	9	656	0.532	7.9
Hospital	4	3306	1.194	22.0
School	8	1115	0.592	11.5
Municipal or Federal	15	3215	0.587	15.6
Lodge, Gym, Hall or Auditorium	12	880	0.390	12.4
Miscellaneous	7	1387	0.479	21.4

[a]Principles of Economical Heating, *National Association of Building Owners and Managers*

building. Since the fuel consumption is proportional to the weather (degree-days) and since the periods to be compared may not have the same weather conditions, the comparison can be made only after the fuel consumptions have been computed on a comparable weather basis, that is, upon the actual number of degree days occurring for a given month and year in the city under consideration. Since fuel consumption is proportional to the number of degree-days, plant operators frequently compute each month the fuel burned per degree-day by the heating plant. The resulting unit figure, by eliminating the outside temperature variable, indicates whether the operating efficiency of the plant is above or below the previous month or year.

The figures in Table 7 illustrate a typical example of a method of using the degree-day for making heating comparisons for one building for two consecutive heating seasons. The heat quantity figures inserted are pounds of steam, but a similar comparison could be made using pounds of coal, gallons of oil, or cubic feet of gas.

For such a comparison, a two-year record is often used, as shown in Table 7. The year under consideration may then be compared, month by month, with the previous year. Column 3, Consumption for Heating, would be used if the same fuel is used for heating and process steam. Some reasonable figure must be assumed for the process requirement and should be deducted from the amount shown in column 2. This would leave in column 3 only the fuel chargeable to heating. The degree-day values in column 5 are obtainable from the local Weather Bureau. Figures in column 6 are obtained by dividing corresponding values in column 3 by the degree days in column 5. The heating index in column 6 is, then, a figure of heat consumption, corrected for outdoor temperature,

TABLE 7. HEAT CONSUMPTION RECORD FOR COMPARISON

Heating Season	Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6	Col. 7
		Total Consumption	Consumption for Heating	Avg Mean Temp.	Deg Days 65 F Base	Lb/Deg Day	Lb/Deg Day/ M Cu Ft
1942-43	Sept.	337,500	170,500	65	146	1,170	0.575
	Oct.	834,200	667,200	53	339	1,966	0.970
	Nov.	1,446,600	1,279,600	44	641	1,990	0.982
	Dec.	2,176,400	2,009,400	25	1,233	1,630	0.804
	Jan.	2,332,200	2,165,200	22	1,297	1,670	0.822
	Feb.	2,131,100	1,964,100	28	1,106	1,775	0.888
	Mar.	2,021,900	1,854,900	31	1,032	1,799	0.885
	Apr.	1,241,500	1,074,500	43	647	1,660	0.818
	May	672,500	505,500	55	303	1,670	0.822
	June	258,600	91,600	......	50	1,830	0.905
	July	188,400	....	......	......	......	......
	Aug.	180,100	....	......	......	......	......
	Total	13,821,000	....	......	......	......	......
1943-44	Sept.	330,200	146,200	61	167	875	0.431
	Oct.	887,100	703,100	52	410	1,718	0.845
	Nov.	1,525,200	1,341,200	39	812	1,653	0.815
	Dec.	2,045,500	1,861,500	28	1,120	1,660	0.817
	Jan.	1,933,400	1,749,400	30	1,044	1,670	0.825
	Feb.	1,990,200	1,806,200	30	1,111	1,624	0.800
	Mar.	1,984,100	1,800,100	31	1,021	1,760	0.868

If, for example, the heat consumption in March, 1943, is compared with that in March, 1944, it will be found that in the latter the steam consumption is 1799 − 1760 = 39 lb less which is a decrease of 2.2 per cent.

and should be relatively constant month by month. Column 7 in Table 7 may be used if the heat consumption is to be compared on a building volume basis with average values shown in Table 6.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are available for a number of buildings in Detroit, as shown in Table 8[6].

These maximum demands were measured by an attachment on the condensation meter and therefore represent the amounts of condensation passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible.

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization. Thus, in Table 8, the theaters, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

SEASONAL EFFICIENCY

The task of predicting fuel consumption within reasonably accurate limits is a simple one where sufficient experience data are available for the fuel in question. Such data can be analyzed to the point where

TABLE 8. BUILDING LOAD FACTORS AND DEMANDS OF SOME DETROIT BUILDINGS

BUILDING CLASSIFICATION	LOAD FACTOR	LB OF DEMAND PER (HOUR) (SQ FT OF EQUIVALENT INSTALLED RADIATOR SURFACE)
Clubs and Lodges	0.318	0.184
Hotels	0.316	0.207
Printing	0.287	0.217
Offices	0.263	0.209
Apartments	0.255	0.225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0.248
Banks	0.203	0.158
Churches	0.158	0.152
Department Stores	0.138	0.145
Theaters	0.126	0.151

average unit factors can be determined and expressed in such terms as, for example, cubic feet of gas actually burned per (square foot of calculated steam radiator surface) (degree-day). The unit U can be inserted directly in Equation 4 without reference to efficiency. Such experience factors are available for gas (see Table 2) and for district steam (Table 6), but not for coal or oil.

Since values of U are not available for oil or coal, an assumed seasonal efficiency E must be used. Selection of a value for this E must be made with caution, for its use implies a meaning not commonly associated with the word efficiency and consequently is frequently misleading.

The input of heat to a building consists not only of the energy in the fuel but that from occupants, the sun, appliances, processes, and all other sources. In many cases these make up, over a period, an important percentage of the total heat required, and if they are not taken into account a calculation of *efficiency* can show a figure over 100 per cent.

For this and other reasons the actual *seasonal efficiency* is a difficult thing to determine. Published data are widely scattered and insufficient. From the available published material it is found that the seasonal efficiency varies over a wide range, depending on the fuel used, and it varies widely even for a given fuel. For example, in a recent survey of 30 houses in one locality there was found a variation of from 45 to 75 per cent in the utilization *efficiency* depending on the fuel [7].

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Gravity Warm Air Systems

Warm Air Leaders, Stacks, and Registers; Return Air Ducts, Grilles, and Shoe Connections; Outline of Design Procedure

WARM air heating systems of the *gravity* type are described in this chapter [1]. In these systems the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the mechanical type a fan may supply all or part of the motive head.

In general, a warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are run in the inside partitions of the building are called stacks. The heated air is finally discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard. A sectional view of a typical plant showing good installation practice is given in Fig. 1.

The air supply to the furnace is usually taken entirely from inside the building through one or more recirculating ducts, although in some cases an outside air supply duct is provided.

WARM AIR LEADERS, STACKS, AND REGISTERS

In a gravity circulating warm-air furnace system, the size of the leader pipe to a given room depends upon the length of the leader and the temperature of the warm air entering the room at the register. For most successful operation, the furnace should be centrally located with respect to register and stack positions so that the leaders will be of uniform length and as short as possible, in which case the frictional resistance to air flow and the temperature loss from the ducts will be about the same for all runs.

In the Standard Code for Installation of Gravity Warm Air Heating Systems, the design was originally based on the heat carrying capacities per square inch of leader pipe area with register air temperatures of 175 F. In a recent revision of the entire design procedure, as shown in the section entitled *Outline of Design Procedure*, the carrying capacities of leader pipes have been expressed directly in terms of Btu per hour.

In general it is advisable to use two or more leader pipes to rooms requiring more than the capacity of a 12 in. round pipe. The tops of all sizes of leader pipes should be cut into the furnace bonnet at the same elevation, and from this point there should be a uniform upgrade of at least 1 in. per foot of run. Leaders over 12 ft in length, or having a large number of elbow fittings should be avoided if possible. In cases where such leaders are necessary, it is recommended that smooth transition fittings be used, and that duct insulation be applied. Asbestos paper, unless of the corrugated type, should not be considered as insulation. To assist in balancing the air distribution of the system, a damper should be placed in each leader pipe except one, this latter leader preferably being connected to a room heated at all times, such as a living room.

In a gravity circulating system, the ratio of stack to leader area is quite important, although little is gained by providing wall stacks with areas in excess of 75 per cent of their connected leader pipe area. In most cases a 3¼ in. × 12 in. stack is the largest which can be installed in normal wall construction. Hence, any room having a heat loss much in excess of 9000 Btu per hour, will require two or more stacks, or one oversized stack built into a 6 in. studding space, providing the design register temperature is to be retained at the value of 175 F as recommended.

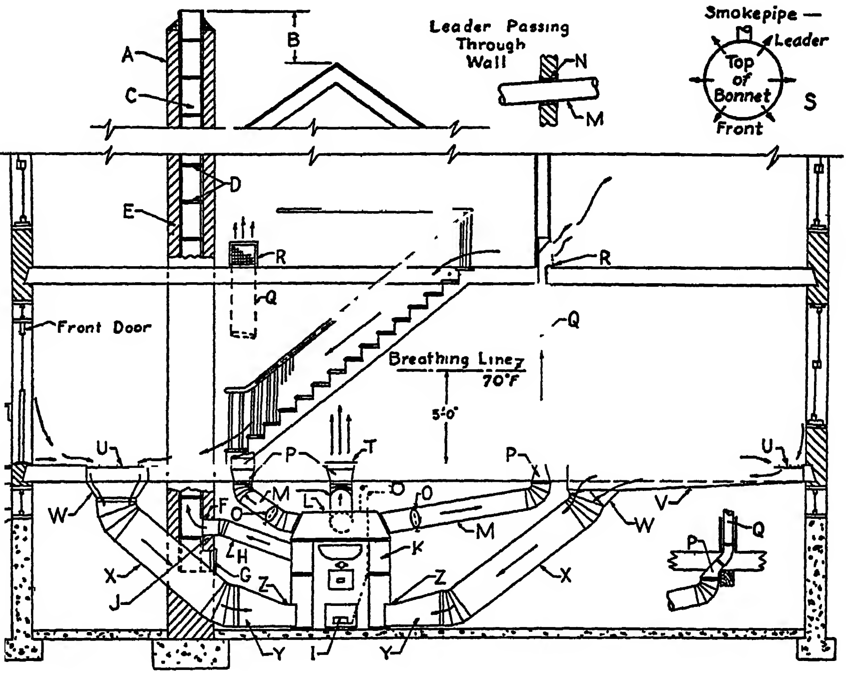


FIG. 1. A SECTIONAL VIEW OF A TYPICAL PLANT SHOWING GOOD INSTALLATION PRACTICE[a]

A. House chimney, no bends nor offsets.
B. Top of chimney at least 2 ft above ridge of roof.
C. Flue lining, fireclay.
D. All joints air tight.
E. At least 8 in. brick.
F. No other connection beside that to furnace
G. Cleanout frame and door, airtight.
H. Smoke pipe, end flush with inner surface of flue.
I. Draft door.
J. Use flue thimble.
K. Casing body.
L. Casing hood or bonnet, top of all leader collars on same level.
M. Round leader, pitch 1 in. per foot.
N. Sleeve with air space around leader where passing through wall.
O. Dampers in all leaders.
P. Transition fittings.
Q. Rectangular wall stack.
R. Baseboard register.
S. Distribute pipes equally around bonnet.
T. Floor register.
U. Return air face.
V. Panning under joist.
W. Transition collar.
X. Round return pipe.
Y. Transition shoe.
Z. Top of shoe at casing not above grate level.

[a]From *N.W.A H.&A.C. Assn.* Standard Code Application Manual.

Registers used for discharging warm air into rooms should have a net area not less than the area of the leader pipe to which the register is attached. First story registers should be connected through boot and register box extensions having areas at least equal to leader areas. Upper story registers should be of the same width as the wall stack, and should be placed either in the baseboard or sidewall, preferably without offsets.

First story registers may be of the baseboard or floor type, with the former location preferred. High sidewall registers in gravity systems deliver more warm air into the room than do baseboard registers, but most of the additional air merely results in high temperatures at the ceiling.

RETURN AIR DUCTS, GRILLES, AND SHOE CONNECTIONS

The ducts through which air is returned to the furnace should be designed to minimize resistance to air flow. They should be of ample area, in excess of the total area of warm-air pipes, and should be streamlined. Horizontal ducts should pitch at least ½ in. per foot downward toward the furnace, avoiding fittings which would require lifting of the return air after the duct has passed under some obstacle.

The return air grilles should have free areas at least equal to the ducts to which they connect and should be installed in the floor, or in the baseboard with the top edge of the grille not more than about 14 in. above the floor line. Frictional resistance in the return air system is as detrimental as is resistance in the warm-air system, so that care should be exercised in locating return air grilles which require long return ducts.

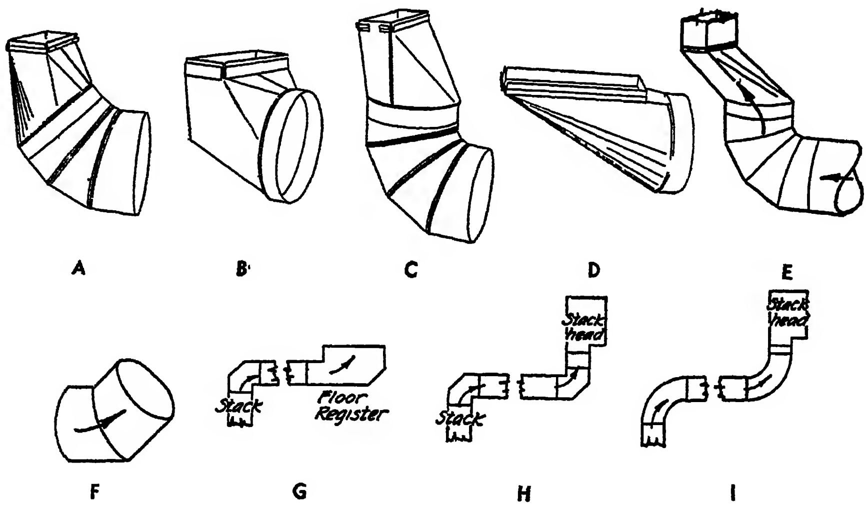


FIG. 2. TYPICAL WARM AIR BOOTS

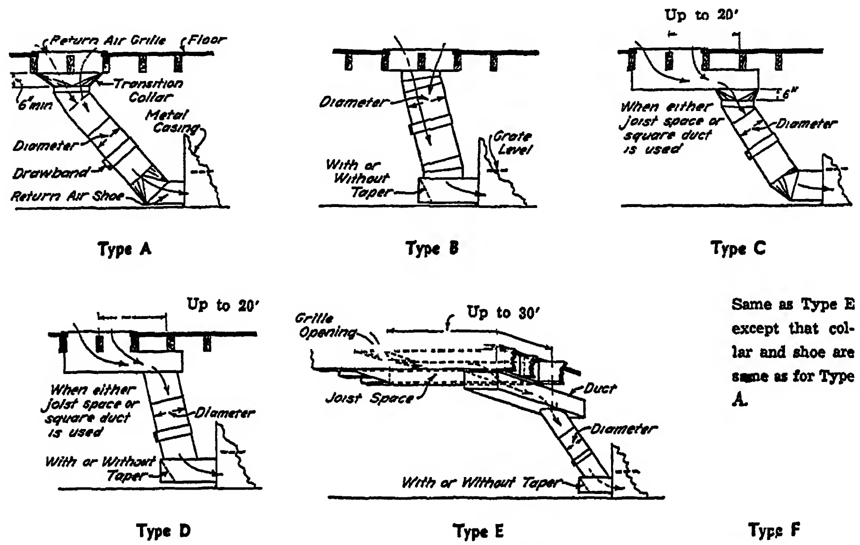
The placement and number of return grilles will depend upon the size, details, and exposure of the house. Small compactly built houses may be adequately served by a single return grille effectively placed in the central hall. It is usually desirable to have two or more returns, provided that in two-story residences one return is placed to effectively receive the return air at the foot of the stairs. A return air connection must be carried to any room whose floor level is below that of adjacent rooms.

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus, in rooms having only small windows the grille can be brought as close to the furnace as possible, but if the room has large window exposure the grille should be located near the exposure. The frictional resistance of the long ducts used in parallel with short return ducts must be reduced to compensate for the length. Return ducts from upstairs rooms may be

necessary in spaces which are closed off from the rest of the house or which have much outdoor exposure. Return grilles on different floor levels should not be connected to the same vertical return duct.

Ducts returning air to the furnace should avoid heat sources which tend to reheat the return air. If the duct must be run over the top of the furnace, or above the vent pipe from the furnace, insulation should be interposed between the heat source and the duct.

Circulation of air is facilitated if the air can slide down a pipe inclined at approximately 45 deg and into a furnace shoe connection having a cross-sectional area equal to that of the pipe. The top of the return shoe should enter the casing below the level of the grate in the case of a coal



Note: For Types C, D, E, and F return-air duct systems, reduce the carrying capacities shown in Table 8 by 1 per cent for each 4 ft additional length in the horizontal run.

FIG. 3. TYPICAL ARRANGEMENTS OF RETURN-AIR DUCT SYSTEMS

furnace, and not more than 14 in. above the floor in the case of oil or gas furnaces. In order to accomplish this the shoe is made wide.

OUTLINE OF DESIGN PROCEDURE

The data underlying the design procedure are given in detail in a circular[2] issued by the University of Illinois. In this procedure the design of the warm-air duct system is considered as an entire unit, so that for a given heat loss the sizes of leaders, stacks, boots, stackheads, and registers are all correlated. Similarly in the case of return ducts, the selection specifies a complete unit consisting of return grille, return duct, and shoe connection.

Recommended Standard Sizes

For the purpose of simplification and standardization, selected combinations of commercial sizes of warm air pipes, return air pipes, ducts, grilles, fittings, and registers are designated as Combination Numbers. The numbers assigned and the combinations selected as standard are listed in the following tables [3].

Table 1—Combination Numbers 1 to 5—First Story Warm Air Ducts and Registers.

Table 2—Combination Numbers 11 to 16—Second Story Warm Air Ducts, Single Wall Stacks, Fittings, and Registers.

Table 3—Combination Numbers 21 to 24—Second Story Warm Air Ducts, Double Wall Stacks, Fittings, and Registers.

Table 4—Combination Numbers 31 to 38—Return Air Ducts, Fittings, and Grilles.

The selected types of boots are shown in Fig. 2 and their resistances expressed in equivalent elbows are shown in Table 5. It is essential that free areas be maintained throughout fittings.

The selected types of return air ducts and fittings are shown in Fig. 3.

TABLE 1. FIRST STORY WARM-AIR DUCTS[a]

Combination No.	Leader Pipe Diameter, In.	Register Size, In		
		Floor	Baseboard	
			Size	Extension
1	8	8 x 10	10 x 8	2¼
2	9	9 x 12	12 x 8	2¼
3	10	10 x 12	12 x 9	3¼
4	12	12 x 14	13 x 11	5¼
5	14	14 x 16	———	——

[a]When the calculations indicate a requirement for a given room greater than Combination No. 4, two or more smaller units totalling the required capacity are recommended.

TABLE 2. SECOND STORY WARM-AIR DUCTS—SINGLE WALL STACKS AND FITTINGS

Combination No.	Leader Pipe Diameter, In.	Stack[b] Size In.	Register Size, In.			
			Floor	Baseboard		Sidewall
				Size	Extension	
11	8	10 x 3¼	8 x 10	10 x 8	2¼	10 x 8
12	9	12 x 3¼	9 x 12	12 x 8	2¼	12 x 8
14	10	14 x 3¼	10 x 12	12 x 8	2¼	12 x 8
15	12	12 x 5¼	———	12 x 9	3¼	———
16	12	14 x 5¼	———	13 x 11	5¼	———

[b]Recommended stack sizes. Tables may also be applied to 3 in. and 3½ in stack depths.

TABLE 3. SECOND STORY WARM-AIR DUCTS—DOUBLE WALL STACKS AND FITTINGS

Combination No.	Leader Pipe Diameter, In.	Stack Size, In.		Register Size, In.			
		Internal	External	Floor	Baseboard		Sidewall
					Size	Extension	
21	8	2½[c] x 10	3⅛[c] x 10⅝	8 x 10	10 x 8	2¼	10 x 8
22	8	3 x 10	3⅝ x 10⅝	8 x 10	10 x 8	2¼	10 x 8
23	9	2½[c] x 12	3⅛[c] x 12⅝	9 x 12	12 x 8	2¼	12 x 8
24	9	3 x 12	3⅝ x 12⅝	9 x 12	12 x 8	2¼	12 x 8

[c]Commercial sizes vary ⅛ in. from values shown.

Carrying Capacity

The Btu carrying capacities of the selected warm air and return air combinations are shown in Tables 6, 7 and 8:

Table 6—Combination Numbers 1 to 5—Warm Air Carrying Capacities to first story registers with 1 to 5 elbows and with leaders 4 to 24 ft long.

Table 7—Combination Numbers 11 to 16 and 21 to 24—Warm Air Carrying Capacities to second story registers with 1 to 5 elbows with leaders 4 to 24 ft long.

Table 8—Combination Numbers 31 to 38—Return Air Carrying Capacities for types *A*, *B*, *C*, *D*, *E*, and *F* return combinations.

TABLE 4. RETURN AIR DUCTS

Combination No.	Duct Dia. In.	Area at Shoe Connection, Sq In.	Metal Grille Sizes: Choose One			When Joist Lining is Used[d]		When Duct is Used: Choose One	
			A	B	C	No. of Joists Lined	Minimum[e] Depth, In.		
31	10			8 x 14	10 x 12	1	7	14 x 6	12 x 8
32	12		6 x 30	8 x 24	12 x 14	1	9	22 x 6	16 x 8
33	14	170	8 x 30	10 x 24	14 x 16	1	12	28 x 6	22 x 8
34	16	220	10 x 30	12 x 24		2	8	28 x 8	22 x 10
35	18	280	12 x 30	14 x 24		2	10	36 x 8	28 x 10
36	20	340	14 x 30	18 x 24		2	12.5	36 x 10	30 x 12
37	22	420	18 x 30			2	15.0	42 x 10	36 x 12
38	24	500	20 x 30			2	18.0	42 x 12	36 x 14

[d]Based on 14 in. space between joists.

[e]Use full depth of joist except when joist depth is less than minimum depth required, *when pan must be used.*

TABLE 5. RESISTANCES OF WARM AIR BOOT COMBINATIONS EXPRESSED IN ELBOW EQUIVALENTS

Warm Air Boot	Name of Combination	Equivalent No. of 90-Deg Elbows
A	45-Deg Angle Boot and 45-Deg Elbow	1
B	90-Deg Angle Boot	1
C	Universal Boot and 90-Deg Elbow	1
D	End Boot	2
E	Offset Boot	2½
F	45-Deg Angle	½
G	Floor Register—Second Story	3
H	Offset	3
I	Offset	2

Design Procedure

The steps to be taken in designing a gravity warm air duct system are:

1. Calculate the heat loss from each room as explained in Chapters 6, 8 and 14.

2. Prepare a layout showing (*a*) furnace, (*b*) chimney connection, (*c*) warm air registers (whether floor, baseboard or wall), (*d*) return air grilles.

3. Indicate on each warm air run (using symbols shown in Fig. 4): (*a*) whether the room to be heated is on the first or second story, (*b*) the approximate length of leader pipe in the basement, (*c*) the number of right angle elbows required, including the elbow at the boot connection (see Fig. 2), (*d*) whether the register is to be located in the floor, in the baseboard, or in the wall.

4. Show the number and proposed locations of return air grilles and the type of return air system (see Fig. 3).

TABLE 6. WARM AIR CARRYING CAPACITY, BTU DELIVERED, FIRST STORY REGISTERS[a]

Length of Leader Pipe—in Feet

Combination No.	No. of Elbows	4 Ft	6 Ft	8 Ft	10 Ft	12 Ft	14 Ft	16 Ft	18 Ft	20 Ft	22 Ft	24 Ft
1	1	6,020	5,850	5,680	5,510	5,340	5,170	5,000	4,830	4,660	4,490	4,320
2		7,620	7,400	7,180	6,970	6,760	6,540	6,320	6,110	5,890	5,680	5,460
3		9,400	9,140	8,870	8,600	8,340	8,070	7,810	7,540	7,270	7,010	6,740
4		13,350	12,970	12,590	12,210	11,830	11,450	11,080	10,700	10,320	9,950	9,560
5		17,520	17,020	16,530	16,040	15,550	15,050	14,550	14,050	13,560	13,060	12,560
1	2	5,850	5,660	5,490	5,330	5,160	5,000	4,840	4,670	4,510	4,340	4,180
2		7,360	7,150	6,940	6,730	6,520	6,320	6,110	5,910	5,700	5,500	5,290
3		9,090	8,840	8,580	8,320	8,060	7,800	7,550	7,290	7,040	6,780	6,520
4		12,910	12,540	12,170	11,800	11,430	11,060	10,690	10,320	9,950	9,580	9,210
5		16,940	16,450	15,990	15,500	15,040	14,550	14,080	13,600	13,120	12,650	12,150
1	3	5,620	5,460	5,310	5,150	4,990	4,830	4,670	4,510	4,350	4,190	4,030
2		7,120	6,910	6,710	6,510	6,310	6,110	5,900	5,700	5,500	5,300	5,100
3		8,780	8,530	8,280	8,030	7,780	7,530	7,290	7,040	6,800	6,550	6,300
4		12,450	12,100	11,750	11,400	11,050	10,700	10,350	10,000	9,650	9,300	8,950
5		16,360	15,900	15,440	14,970	14,510	14,050	13,600	13,130	12,660	12,200	11,750
1	4	5,420	5,260	5,110	4,960	4,800	4,650	4,500	4,350	4,190	4,040	3,890
2		6,860	6,660	6,460	6,270	6,080	5,890	5,690	5,500	5,300	5,110	4,910
3		8,460	8,200	7,980	7,740	7,500	7,260	7,020	6,780	6,550	6,310	6,070
4		12,010	11,670	11,330	10,990	10,650	10,310	9,970	9,630	9,290	8,950	8,610
5		15,770	15,320	14,880	14,420	13,990	13,540	13,100	12,650	12,200	11,750	11,310
1	5	5,240	5,090	4,940	4,790	4,640	4,500	4,350	4,200	4,050	3,910	3,760
2		6,630	6,440	6,250	6,060	5,880	5,690	5,500	5,320	5,130	4,940	4,750
3		8,180	7,950	7,720	7,490	7,260	7,030	6,800	6,560	6,330	6,100	5,860
4		11,610	11,290	10,950	10,620	10,300	9,970	9,640	9,320	8,990	8,660	8,320
5		15,250	14,800	14,380	13,950	13,520	13,090	12,650	12,230	11,800	11,370	10,940

[a]Additional values for 6 and 7 elbows are given in original Manual.

TABLE 7. WARM AIR CARRYING CAPACITY, BTU DELIVERED, SECOND STORY REGISTERS[a]

Length of Leader Pipe—in Feet

COMBINATION No.[b,c]	NO. OF ELBOWS	4 FT	6 FT	8 FT	10 FT	12 FT	14 FT	16 FT	18 FT	20 FT	22 FT	24 FT
11-22	1	8,370	8,140	7,900	7,670	7,430	7,190	6,950	6,710	6,470	6,240	6,000
12-24		10,040	9,760	9,470	9,190	8,900	8,620	8,330	8,050	7,770	7,480	7,200
14		11,710	11,380	11,050	10,720	10,390	10,060	9,720	9,390	9,060	8,730	8,400
15		16,200	15,750	15,300	14,840	14,380	13,920	13,460	13,000	12,550	12,100	11,640
16		18,920	18,390	17,850	17,310	16,780	16,240	15,710	15,180	14,640	14,100	13,570
11-22	2	7,940	7,720	7,500	7,280	7,050	6,830	6,600	6,370	6,150	5,930	5,700
12-24		9,540	9,270	9,000	8,730	8,460	8,190	7,920	7,650	7,380	7,110	6,840
14		11,120	10,810	10,500	10,180	9,870	9,550	9,230	8,920	8,610	8,290	7,980
15		15,400	14,970	14,530	14,100	13,670	13,230	12,800	12,360	11,930	11,500	11,070
16		17,980	17,470	16,960	16,450	15,950	15,430	14,830	14,420	13,910	13,400	12,890
11-22	3	7,530	7,320	7,110	6,900	6,680	6,470	6,250	6,040	5,830	5,620	5,400
12-24		9,030	8,780	8,520	8,270	8,010	7,750	7,500	7,240	6,990	6,730	6,470
14		10,530	10,240	9,940	9,650	9,350	9,050	8,750	8,450	8,160	7,860	7,560
15		14,580	14,180	13,780	13,370	12,950	12,530	12,120	11,710	11,300	10,890	10,480
16		17,040	16,550	16,070	15,580	15,110	14,620	14,140	13,660	13,180	12,700	12,210
11-22	4	7,120	6,920	6,720	6,520	6,310	6,110	5,900	5,700	5,500	5,300	5,100
12-24		8,530	8,290	8,050	7,810	7,570	7,330	7,080	6,840	6,600	6,360	6,120
14		9,950	9,670	9,390	9,110	8,830	8,550	8,260	7,980	7,700	7,420	7,140
15		13,780	13,390	13,000	12,610	12,220	11,830	11,440	11,050	10,670	10,280	9,890
16		16,080	15,620	15,170	14,710	14,260	13,810	13,350	12,900	12,440	11,980	11,530
11-22	5	6,700	6,510	6,320	6,130	5,940	5,750	5,560	5,370	5,180	4,990	4,800
12-24		8,040	7,810	7,580	7,350	7,130	6,900	6,670	6,440	6,220	5,990	5,760
14		9,370	9,110	8,850	8,580	8,310	8,050	7,780	7,510	7,250	6,980	6,720
15		12,970	12,600	12,240	11,870	11,500	11,140	10,770	10,400	10,040	9,680	9,310
16		15,140	14,710	14,280	13,850	13,420	13,000	12,570	12,140	11,710	11,280	10,850

[a]When floor registers are used, see Fig. 2.

[b]No 21 for Btu values multiply 11-22 values by 0.83.

[c]No. 23 for Btu values multiply 12-24 values by 0 83

5. From Table 6, for first story, or from Table 7 for second story, select the combination number for the warm air system which will supply the heat required to each room, with the number of elbows and length of leader pipe previously determined. Then, using the combination number as found, read directly in Tables 1, 2, or 3 the leader, stack, and register sizes required.

6. From Table 8 select the combination number for the return air system to correspond with the Btu serviced and the type of return air system. Then from Table 4 select the duct and grille sizes, etc., corresponding to the same combination number.

7. Select a furnace having a register delivery, in Btu per hour, equal to the total heat loss from the structure.

TABLE 8. RETURN AIR—CARRYING CAPACITY—BTU SERVICED

Return Air Combination No.	Duct Dia In.	Type A Btu per Hr	Types B and C Btu per Hr	Type D Btu per Hr	Type E Btu per Hr	Type F Btu per Hr	Return Air Combination No.
31	10	11,300	9,500	7,800	5,000	7,800	31
32	12	16,300	13,700	11,300	7,200	11,300	32
33	14	22,200	18,700	15,300	9,800	15,300	33
34	16	29,000	24,400	20,000	12,800	20,000	34
35	18	36,700	30,800	25,300	16,200	25,300	35
36	20	45,300	38,000	31,300	20,000	31,300	36
37	22	54,800	46,000	37,800	24,100	37,800	37
38	24	65,200	54,800	45,000	28,700	45,000	38

Examples 1 and 2 will illustrate the use of the tables in selecting warm air and return system sizes.

Example 1. For a room which has a heat loss of 22,500 Btu per hour select the size of first story warm air system. There are three elbows and the leader is approximately 10 ft long.

Solution: Since 22,500 Btu is beyond the capacities shown in Table 6, it is necessary to select two units of 11,250 each. From Table 6 in 10 ft leader column and in section for three elbows find 11,400 as nearest capacity which corresponds to Combination Number 4 in first column. Refer to Combination Number 4 in Table 1 and find that the leader should be 12 in. in diameter and should be used with a 12 x 14 in. floor register or a 13 x 11 in. baseboard register with a 5¼ in. extension.

Example 2. What is the size of a return system of Type D which is to service 35,000 Btu per hour?

Solution: From Table 8 find Combination Number 37 which will service 37,800 Btu per hour. Refer to Table 4 to find that Combination Number 37 will require a 22-in. diameter duct, a shoe area of 420 sq in., a metal grille 18 x 30 in., a duct 42 x 10 in. or 36 x 12 in. If joist lining is used the minimum depth should be 15 in. for two 2-joist spaces 14 in. wide, or 10 in. for three joist spaces.

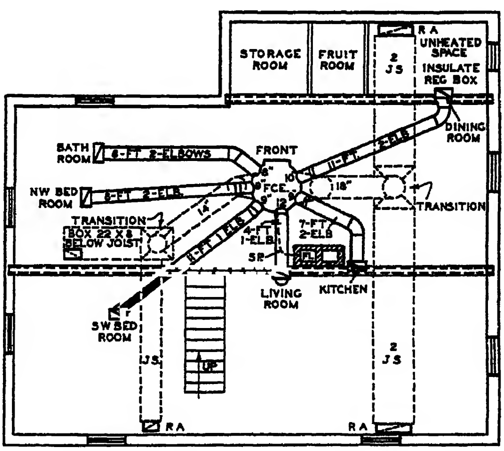


FIG. 4. TYPICAL BASEMENT LINE DRAWING

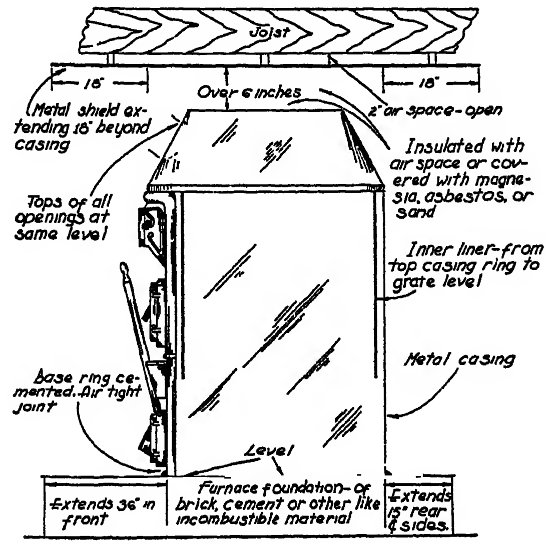


FIG. 5. DETAILS OF FURNACE BONNET, CASING, AND FOUNDATION (FROM GRAVITY CODE AND MANUAL)

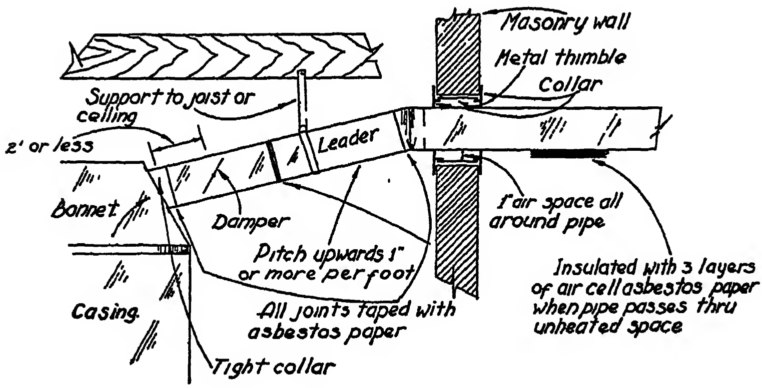


FIG. 6. DETAILS OF BONNET AND LEADER OF GRAVITY WARM-AIR FURNACE (FROM GRAVITY CODE AND MANUAL)

Figs. 5 and 6 show recommended practice as given in the *N.W.A.H. &A.C. Assn.* Gravity Code and Manual. For construction, design features, and ratings of gravity furnaces see Chapter 18.

REFERENCES

1—The engineering data were obtained from University of Illinois, *Engineering Experiment Station Bulletins* Nos. 141, 188, 189 and 246; Warm Air Furnaces and Heating Systems, by A. C. Willard, A. P. Kratz, V. S. Day, and S. Konzo. See also Gravity Code and Manual for Gravity Warm Air Heating Systems, published by the *National Warm Air Heating and Air Conditioning Association.*

2—Simplified Procedure for Selecting Capacities of Duct Systems for Gravity Warm-Air Heating Plants, by A. P. Kratz and S. Konzo (*University of Illinois, Engineering Experiment Station Circular* 45, Dec., 1942)

3—Gravity Code and Manual for Gravity Warm Air Heating Systems, Second Edition, 1945, *National Warm Air Heating and Air Conditioning Association.*

CHAPTER 22

Mechanical Warm Air Systems

Air Distribution, Standard Combinations of Parts, Automatic Controls, Simplified Design of Heating System, Cooling Methods, Cooling System Design

IN mechanical warm air or fan furnace heating systems [1], the air circulation is effected by motor-driven fans instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom, as in gravity systems described in Chapter 21. The advantages of mechanical systems, as compared with gravity systems, are:

1. The furnace need not be centrally located but may be placed in any part of the basement.

2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view where desired.

3. Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.

4. Humidity control is more readily attained.

5. The air may be cleaned by sprays or filters, or both.

6. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.

7. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

8. Ventilation air may be positively introduced and heated.

The construction features of mechanical warm air furnace units and discussions of the function and selection of the various parts, such as the furnace, casings, motors, filters and controls are included in Chapter 18.

AIR DISTRIBUTION

The conditions of comfort obtained in a room are greatly influenced by the type of register used and the locations of the supply registers and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply register affect the room conditions much more than the changes in the location of the return grilles. One method is to locate the supply register near the floor so that the warm air from the register *blankets* a cold wall, and mixes with the cold air dropping off from the exposed walls and glass. Another method is to locate the supply openings near the floor on the inside wall and the return openings near the greatest outside exposure. In any case the warm air registers should be located so that the air stream never discharges directly into space that will normally be occupied by people at rest. Tests in the Warm Air Research Residence [2] have indicated that continuous fan operation provided better results than intermittent fan operation.

Register and Grille Openings

Supply registers located in the floor require attention to keep them clean and are usually avoided. Tests conducted in the Warm Air Research

Residence have indicated that comparable results are obtainable with either high side wall or baseboard registers, if proper registers and air velocities are selected. Baseboard registers should be of a deflecting-diffuser type which throw the air downward toward the floor and diffuse it at the same time. For baseboard registers air temperatures under 125 F and air velocities over 500 fpm should be avoided as they may cause drafts.

High side wall registers must be of such type that the air is delivered horizontally or in a slightly downward direction, and must be so located as to avoid impingement of air on ceiling or wall. Directional flow diffusing type should be used to insure best results. Register air velocities

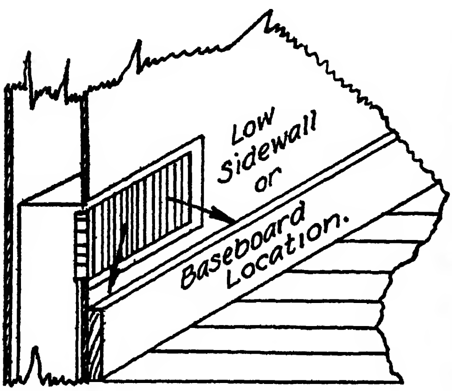


FIG. 1. RECOMMENDED TYPE OF BASEBOARD AND LOW SIDEWALL REGISTERS[a]

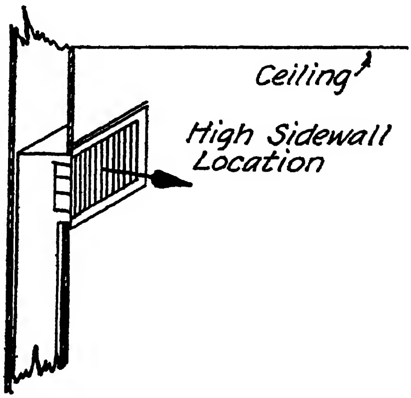


FIG. 2. RECOMMENDED TYPE OF HIGH SIDEWALL REGISTERS[b]

[a]Vertical bars with adjustable deflection, or fixed vertical bars with deflections to right and left not exceeding about 22 deg. For low sidewall location, the deflection for horizontal, multiple valve, registers should not exceed 22 deg. For baseboard locations, the deflection for horizontal, multiple valve, registers should not exceed about 10 deg.

[b]Horizontal valves, in back or front, to give downward deflections not to exceed from 15 to 22 deg.

should be such that the air stream carries to the opposite exposure. Velocities under 500 fpm are not recommended. Basic rules for the location and selection of registers are given in Section C of the Code and Manual (Textbook Section 7) of the *National Warm Air Heating and Air Conditioning Association.*

Registers should be well proportioned and decorated to harmonize with the trim. Air supply registers should be equipped with dampers and all registers should be sealed against leakage around edges. The register types shown in Figs. 1 and 2 have been recommended as standard by the *National Warm Air Heating and Air Conditioning Association.*

Velocities through registers may be reduced by the use of registers larger than the connecting ducts. Diffusers should be used to spread the air uniformly over the register face.

Return air grilles may be located in hallways, near entrance doors, under windows, in exposed corners, or inside walls, depending on location of supply registers. Baseboard returns are preferable to floor grilles.

Dampers

Suitable dampers for air direction or volume control are essential to any trunk or individual duct system. Special care must be used in the design

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10-year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.

7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained.

8. In the selection of cooling coils, the additional frictional resistance of the coil to flow of air must be given consideration.

9. Cooling the structure by introducing large quantities of air from outdoors at night tended to reduce the amount of cooling required on the following day and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

METHOD OF DESIGNING COOLING SYSTEM

The general procedure which may be used for the design of a summer cooling system in a forced-air installation is:

1. Calculate heat gain for each room or space to be conditioned. (See Chapters 6 and 15.) Allowance for addition of outside air must be included in this calculation.

2. Select a temperature of air leaving supply inlets. In Research Residence tests a value of from 65 to 70 F was found satisfactory.

3. Determine indoor conditions to be maintained. In Research Residence 80 F dry-bulb and 45 per cent relative humidity were found satisfactory.

4. Determine the quantity of air to be introduced into each room. (See Chapter 43.)

5. Estimate heat gain in duct system between cooling unit and supply registers.

6. Calculate the sensible and latent heat to be removed by the cooling unit.

7. Determine size of ducts in duct system and size of registers, as explained in Chapters 40 and 41.

8. Determine pressure loss in duct system and select fan as also explained in the same section.

9. Select cooling unit from manufacturer's data. Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus.

10. Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing. (See Chapter 25.)

11. If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation. Adjustable dampers will be necessary to make changes in air distribution for the two seasons. Provision must also be made for changing fan speeds for summer and winter operation.

REFERENCES

[1]—Specifications for the furnace unit and the installed duct system are shown in The Yardstick (Textbook Section 8) and the Code and Manual (Textbook Section 7) published by *National Warm Air Heating and Air Conditioning Association.*

[2]—Performance of a Forced Warm-Air Heating System as Affected by Changes in Volume and Temperature of Air Recirculated, by A. P. Kratz and S. Konzo (A.S.H.V E. TRANSACTIONS, Vol. 48, 1942, p 393).

[3]—Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 37).

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TABLE 1. WARM AIR DUCT SYSTEM COMBINATIONS OF PARTS SELECTED AS STANDARD

Combination No.	Stack Size In.	Branch Pipe Size, In.		Register Size, In. (See Figs. 1 and 2)				Required Increase in Width of Trunk Duct, In.
		Round	Rectangular	Baseboard or Low Sidewall	High Sidewall: For throws less than 13 ft	High Sidewall: For throws more than 13 ft	Floor Registers[a]	
1	2	3	4	5	6	7	8	9
41	10 x 3¼	6	4 x 8	10 x 6	10 x 6	10 x 4	8 x 10[a]	1
42	10 x 3¼	6	4 x 8	10 x 6	10 x 6	10 x 4	8 x 10[a]	2
43	12 x 3¼	7	5 x 8	12 x 6	12 x 6	12 x 4	9 x 12[a]	3
44	14 x 3¼	8	6 x 8	14 x 6	14 x 6	14 x 4	9 x 12[a] or longer	4
45	10 x 3¼ (2-Stacks)	9	8 x 8	(2) 10 x 6 or (1) 24 x 6	(2) 10 x 6 or (1) 24 x 6	(2) 10 x 4 or (1) 24 x 4	10 x 12[a]	6
46	12 x 3¼ (2-Stacks)	10	10 x 8	(2) 12 x 6 or (1) 30 x 6	(2) 12 x 6 or (1) 30 x 6	(2) 12 x 4 or (1) 30 x 4	12 x 14[a]	7

[a]Use these items only when the building construction or capacity requirements necessitate the use of floor registers. The sizes listed for floor registers correspond to the standard sizes for gravity warm-air systems, except for the sizes of the floor box collars. The use of standard blind boxes is suggested.

SIMPLIFIED METHOD OF DESIGNING DOMESTIC FORCED-AIR SYSTEMS

A simplified method for selecting the combinations of branches, boots, stacks, and registers, is given in the Code and Manual (Textbook section No. 7) of the *National Warm Air Heating and Air Conditioning Association*. In this method the sizes of the branch ducts are obtained from two tables giving their Btu capacities. The proper combination of parts for each branch can be determined if the following information is available.

a. Location of room, that is, whether on first or second story.

b. Actual length of basement duct from bonnet to boot, in feet.

c. Btu loss from room to be heated.

d. Equivalent lengths in feet of all fittings and of the register. Fig. 3 shows the values of equivalent lengths of fittings commonly used for domestic systems.

This simplified method is applicable to structures having heat losses not in excess of approximately 150,000 Btu per hour. The capacities shown in Tables 3 and 4 are based upon the most reliable data pertaining to friction losses and temperature drops in ducts. They are also based upon a 100 deg temperature rise of the air, and a static pressure available for overcoming friction losses in the external duct system alone of 0.20 in. water gage. The use of this method assumes that the fan in the fan-furnace assembly will be capable not only of overcoming the resistance of the external duct system alone, but also the resistances imposed by the blower inlet, the filter, and the furnace casing.

TABLE 2. RETURN-AIR DUCT SYSTEM COMBINATIONS OF PARTS SELECTED AS STANDARD

Combination No	Return-Air Intake Size, In.		Riser Size, In. Where Stack is Used in Stud Space	Branch Pipe Size, In.		When Joist Lining is Used[b] Number of Joist Spaces Lined and Minimum Depth of Space Required	Required Increase in Width of Trunk Duct (for 8 In. Depth of duct), In.
	Baseboard	Floor[a]		Round	Rectangular		
1	2	3	4	5	6	7	8
51	10 x 6	6 x 10 or 4 x 14	10 x 3¼[c]	6	4 x 8	1 space of 3 in. depth	1
52	10 x 6	6 x 10 or 4 x 14	10 x 3¼[c]	6	4 x 8	1 space of 3 in. depth	2
53	12 x 6	6 x 12 or 6 x 14	12 x 3¼[d]	7	5 x 8	1 space of 4 in. depth	3
54	14 x 6	6 x 14	14 x 3¼[d]	8	6 x 8	1 space of 5 in. depth	4
55	24 x 6 or 30 x 6	6 x 30	Two stacks each 10 x 3¼[c]	9	8 x 8	1 space of 6 in. depth or 2 spaces of 3 in. depth	6
56	30 x 6	6 x 30	Two stacks 12 x 3¼[d]	10	10 x 8	1 space of 7 in. depth or 2 spaces of 4 in. depth	7
57	——	8 x 30	——	12	15 x 8	1 space of 9 in. depth or 2 spaces of 5 in. depth	11

[a]Use these items only when building construction, or capacities, requires the use of floor intakes. The sizes listed correspond to standard sizes for gravity installations, except floor box collars. The use of standard blind boxes is suggested.

[b]Based on 14 in. space between joists. Use full depth of joist, except when joist depth is less than minimum depth required, in which case a drop pan must be used. This may occur when two or more return ducts are connected to the same joist space.

[c]If it is desired to use 14 in. x 3⅝ in. stud space, it makes no difference whether this space has protruding keys or not

[d]If it is desired to use 14 in. x 3⅝ in. stud space, the plaster base must be smooth, without any protruding plaster keys to interfere with the flow of air.

The combination numbers shown in the right hand column of Tables 3 and 4 correspond to those given in Tables 1 and 2.

Tables 3 and 4 are also applicable for the selection of the return air branches. A depth of 8 in. has been adopted as the standard for the trunk ducts. The width of a trunk duct serving two branches is determined by adding to the width of the remote branch the value shown in column 9 of Table 1, or column 8 of Table 2.

For buildings having a heat loss in excess of 150,000 Btu per hour the design procedure may be that given in the Technical Code, Fourth Edition, of the *National Warm Air Heating and Air Conditioning Association.* Air duct sizes and air distribution may also be calculated in accordance with data given in Chapters 40 and 41.

COOLING METHODS

A slight cooling effect may be obtained under certain conditions by the use of the cooler basement air. A more positive cooling effect may be

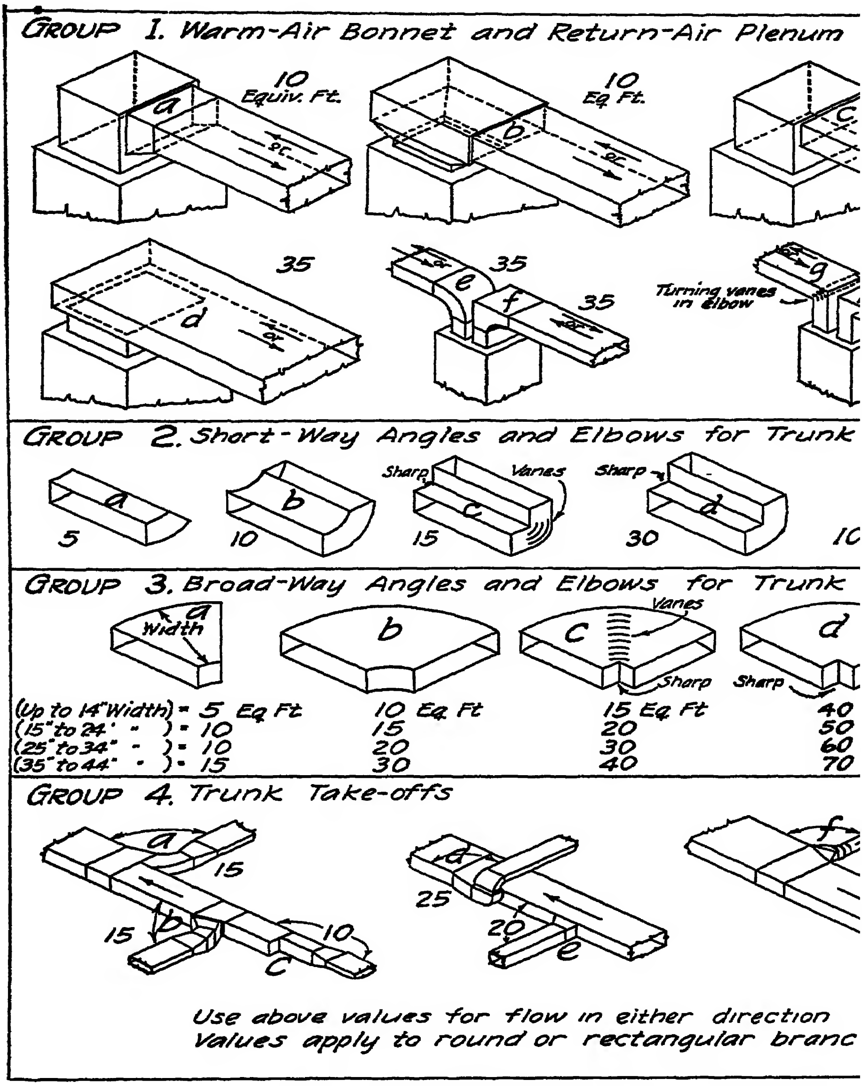
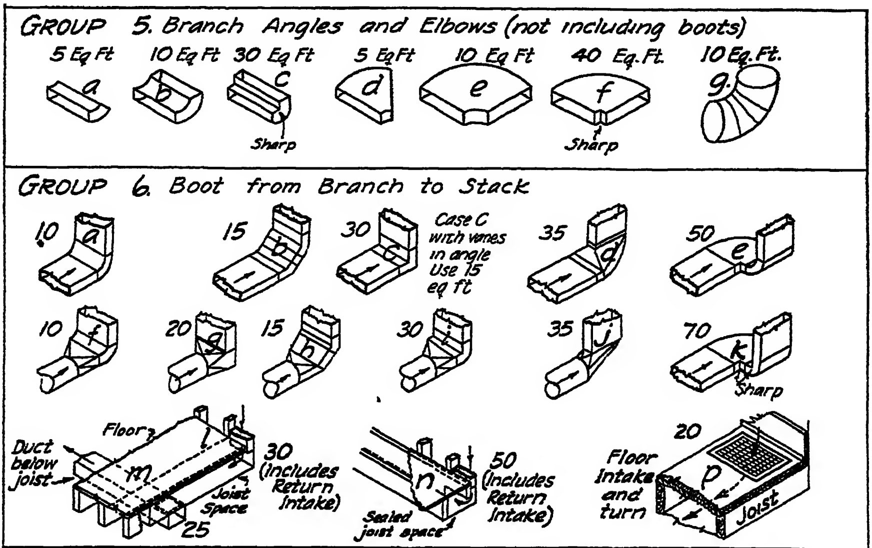


FIG. 3. EQUIVALENT LENGTH OF FITTINGS

obtained by the use of an air washer where the temperatı well water is sufficiently low (55 F or lower), and wl volume of water can be provided. Unless the temperatu water is below the dew-point temperature of the indoor the washer is started, both the relative and absolute hı somewhat increased.

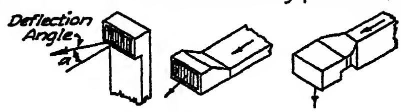
Coils of copper finned tubing through which cold wate available for cooling. They require less space than air w the advantage that no moisture is added to the air when of the water rises above the dew-point. Ample coil ; capacity are necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refri nection with a warm air system and to cool the building



GROUP 7. Registers (Including losses in stackhead and velocity pressure)

TABLE 7.
Equivalent Length for Registers.
(See diagrams at right)



Deflection Angle a	Baseboard or Low Sidewall Register	High Sidewall Register For throw of air of: less than 13 feet	High Sidewall Register For throw of air of: over 13 feet
0°	35 Eq Ft	35 Eq.Ft.	70 Eq Ft
15°	40 " "	40 " "	80 " "
22°	45 " "	45 " "	90 " "
30°	60 " "	60 " "	115 " "
45°	115 " "	115 " "	230 " "

Notes for Table 7

a) When turning vanes are used in stack-head use 0.7 of values in columns 2 to 4

b) Values shown in 2nd and 3rd columns in Table are for 6 in height registers (Standard height) and those in 4th column are for 4 in height registers
For 5 in height registers, multiply values in Col 2 and 3 by 1.3
For 8 in. height registers, multiply values in Col. 2 and 3 by 0.8

For 2-way deflection registers, add the vertical and horizontal deflection angles together and multiply by 0.7. Select closest angle "a" in Col 1 of Table 7

FIG. 3. EQUIVALENT LENGTH OF FITTINGS (Continued)

provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. (See also Chapters 38, 39 and 43.)

Conclusions drawn from studies[4] conducted in the University of Illinois Research Residence, subject to the limitations of the test are:

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hr on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

Table 3. Capacity Tables for Warm-Air and Return-Air Branches[a,b]

FIRST STORY

For Uninsulated Metal Ducts	Actual Length (from bonnet to boot) or (from return plenum to boot) in feet								Warm Air Combination No.	Return Air Combination No.
	1 to 7 ft	8 to 12 ft	13 to 17 ft	18 to 24 ft	25 to 34 ft	35 to 44 ft	45 to 54 ft	55 to 64 ft		
	Col. A	Col. B	Col. C	Col. D	Col. E	Col. F	Col. G	Col. H		
Section A.	7200	6700	6100	5600	4800	4100	3500	3000	41	51
	12500	11700	10800	9900	8500	7400	6400	5500	42	52
40 to 69	16000	15000	14000	13000	11300	9900	8700	7700	43	53
Equivalent Ft for	19100	18000	17000	16000	14200	12500	11000	9600	44	54
Fittings	25000	23400	21600	19800	17000	14800	12800	11000	45[c]	55
and	32000	30000	28000	26000	22600	19800	17400	15400	46[c]	56
Register	80000	75000	70000	65000	56500	49500	43500	38500	....	57[c]
Section B.	5500	5100	4800	4500	3900	3400	3000	2600	41	51
	9900	9200	8600	8100	7100	6200	5400	4600	42	52
70 to 99	13100	12300	11600	10900	9700	8500	7500	6500	43	53
Equivalent Ft	16300	15400	14500	13700	12200	10800	9500	8300	44	54
	19800	18400	17200	16200	14200	12400	10800	9200	45[c]	55
	26200	24600	23200	21800	19400	17000	15000	13000	46[c]	56
	65500	61500	58000	54500	48500	42500	37500	32500	....	57[c]
Section C.	4400	4200	3900	3700	3200	2900	2600	2300	41	51
	8100	7600	7100	6600	5800	5100	4500	4000	42	52
100 to 139	10800	10100	9600	9000	8000	7100	6200	5400	43	53
Equivalent Ft	13700	12900	12100	11300	10000	8900	7900	7000	44	54
	16200	15200	14200	13200	11600	10200	9000	8000	45[c]	55
	21600	20200	19200	18000	16000	14200	12400	10800	46[c]	56
	54000	50500	48000	45000	40000	35500	31000	27000	....	57[c]
Section D.	3900	3700	3500	3300	2900	2500	2200	2000	41	51
	6900	6500	6100	5700	5100	4500	4000	3600	42	52
140 to 189	9300	8700	8100	7600	6800	6000	5300	4700	43	53
Equivalent Ft	11600	11000	10400	9800	8700	7700	6800	6000	44	54
	13800	13000	12200	11400	10200	9000	8000	7200	45[c]	55
	18600	17400	16200	15200	13600	12000	10600	9400	46[c]	56
	46500	43500	40500	38000	34000	30000	26500	23500	....	57[c]
Section E.	3300	3100	2900	2700	2500	2200	1900	1700	41	51
	5700	5400	5100	4800	4300	3800	3400	3000	42	52
190 to 250	7800	7300	6900	6500	5700	5000	4400	3900	43	53
Equivalent Ft	9700	9100	8600	8100	7300	6500	5800	5100	44	54
	11400	10800	10200	9600	8600	7600	6800	6000	45[c]	55
	15600	14600	13800	13000	11400	10000	8800	7800	46[c]	56
	39000	36500	34500	32500	28500	25000	22000	19500	....	57[c]

For Insulated Ducts	Col. A	Col. B	Col. C	Col. D	Col. E	Col. F	
	1 to 9 ft	10 to 17 ft	18 to 24 ft	25 to 34 ft	35 to 54 ft	55 to 74 ft	For ducts that are completely insulated with ½ in. thick insulation from Bonnet to Boot, use these column headings.

[a]These tables are for use in sizing both the warm air and the return air branches.

[b]Frictional resistances and temperature drops in ducts have both been accounted for in these tables.

[c]Use these items only when the building construction, or capacity requirements, necessitates the use of two adjoining stacks or floor registers.

TABLE 4. CAPACITY TABLES FOR WARM-AIR AND RETURN AIR BRANCHES[a,b]

SECOND STORY

FOR UNINSULATED METAL DUCTS	ACTUAL LENGTH (FROM BONNET TO BOOT) OR, (FROM RETURN PLENUM TO BOOT) IN FEET								WARM AIR COMBINATION No.	RETURN AIR COMBINATION No.
	1 TO 7 FT	8 TO 12 FT	13 TO 17 FT	18 TO 24 FT	25 TO 34 FT	35 TO 44 FT	45 TO 54 FT	55 TO 64 FT		
	COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	COL. G	COL. H		
Section A.	6300	5700	5200	4800	4100	3500	3100	2800	41	51
	10900	10000	9200	8500	7300	6400	5600	5000	42	52
40 to 69	14000	13000	12100	11400	10000	8800	7800	6800	43	53
Equivalent Ft for	17000	15900	14900	13900	12400	11100	9900	9000	44	54
Fittings	21800	20000	18400	17000	14600	12800	11200	10000	45[c]	55
and	28000	26000	24200	22400	20000	17600	15600	13600	46[c]	56
Register	70000	65000	60500	57000	50000	44000	39000	34000	....	57[c]
Section B.	5000	4600	4300	4000	3400	3000	2700	2300	41	51
	9000	8200	7600	7100	6200	5400	4700	4200	42	52
70 to 99	11900	11000	10300	9600	8400	7500	6700	6000	43	53
Equivalent Ft	14800	13900	13000	12200	10800	9600	8500	7500	44	54
	18000	16400	15200	14200	12400	10800	9400	8400	45[c]	55
	23900	22000	20600	19200	16800	15000	13400	12000	46[c]	56
	59500	55000	51500	48000	42000	37500	33500	30000	—	57[c]
Section C.	4100	3800	3600	3400	2900	2600	2300	2000	41	51
	7400	6900	6400	6000	5200	4600	4000	3600	42	52
100 to 139	10000	9300	8700	8100	7100	6200	5500	4800	43	53
Equivalent Ft	12500	11600	10800	10100	9000	8000	7200	6500	44	54
	14800	13800	12800	12000	10400	9200	8000	7200	45[c]	55
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	50000	46500	43500	40500	35500	31000	27500	24000	—	57[c]
Section D.	3700	3400	3100	2900	2600	2200	2000	1800	41	51
	6500	6000	5600	5300	4700	4100	3700	3300	42	52
140 to 189	8600	8000	7500	7000	6100	5400	4800	4300	43	53
Equivalent Ft	10900	10100	9400	8800	7800	6900	6100	5400	44	54
	13000	12000	11200	10600	9400	8200	7400	6600	45[c]	55
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	43000	40000	37500	35000	30500	27000	24000	21500	....	57[c]
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190 to 250	7300	6800	6300	5900	5100	4500	3900	3500	43	53
Equivalent Ft	9100	8500	7900	7500	6600	5900	5200	4700	44	54
	11000	10400	9600	8800	7800	6800	6200	5600	45[c]	55
	14600	13600	12600	11800	10200	9000	7800	7000	46[c]	56
	36500	34000	31500	29500	25500	22500	19500	17500	—	57[c]

FOR INSULATED DUCTS	COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	COL. G	FOR DUCTS THAT ARE COMPLETELY INSULATED WITH ½ IN THICK INSULATION FROM BONNET TO BOOT USE THESE COLUMN HEADINGS.
	1 TO 8 FT	9 TO 14 FT	15 TO 20 FT	21 TO 27 FT	28 TO 42 FT	43 TO 54 FT	55 TO 70 FT	

[a]These tables are to use in sizing both the warm air and the return air branches.

[b]Frictional resistance and temperature drops in ducts have both been accounted for in these tables.

[c]Use these items only when the building construction, or capacity requirements, necessitates the use of two adjoining stacks or floor registers.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10-year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.

7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained.

8. In the selection of cooling coils, the additional frictional resistance of the coil to flow of air must be given consideration.

9. Cooling the structure by introducing large quantities of air from outdoors at night tended to reduce the amount of cooling required on the following day and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

METHOD OF DESIGNING COOLING SYSTEM

The general procedure which may be used for the design of a summer cooling system in a forced-air installation is:

1. Calculate heat gain for each room or space to be conditioned. (See Chapters 6 and 15.) Allowance for addition of outside air must be included in this calculation.

2. Select a temperature of air leaving supply inlets. In Research Residence tests a value of from 65 to 70 F was found satisfactory.

3. Determine indoor conditions to be maintained. In Research Residence 80 F dry-bulb and 45 per cent relative humidity were found satisfactory.

4. Determine the quantity of air to be introduced into each room. (See Chapter 43.)

5. Estimate heat gain in duct system between cooling unit and supply registers.

6. Calculate the sensible and latent heat to be removed by the cooling unit.

7. Determine size of ducts in duct system and size of registers, as explained in Chapters 40 and 41.

8. Determine pressure loss in duct system and select fan as also explained in the same section.

9. Select cooling unit from manufacturer's data. Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus.

10. Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing. (See Chapter 25.)

11. If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation. Adjustable dampers will be necessary to make changes in air distribution for the two seasons. Provision must also be made for changing fan speeds for summer and winter operation.

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[3]—Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 37).

[4]—Summer Cooling in the Research Residence, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L Broderick (University of Illinois *Engineering Experiment Station Bulletins* Nos. 290, 305 and 321). A.S.H.V.E RESEARCH REPORT No. 1177—Summer Cooling in the Research Residence with a Gas-Fired Dehydration Cooling Unit, by A. P Kratz, S Konzo and E L. Broderick (A.S.H.V.E TRANSACTIONS, Vol. 47, 1941, p. 203).

CHAPTER 23

Steam Heating Systems and Piping

Classification, Piping for Steam Heating Systems, Steam Flow, Pipe Sizes, Indirect Heating Units, Types of Heating Systems, High Pressure Steam Systems, Boiler Connections, Condensate Return Pumps, Vacuum Pumps, Traps, Drips, Connections to Heating Units, Control Valves

STEAM heating systems may be classified according to any one of, or combination of, the following features: (1) the piping arrangement, (2) the method of returning the condensate to the boiler, (3) the accessories used, (4) the method of expelling or removing the air from the system, (5) the type of control used, and (6) the pressure or vacuum conditions obtained in operation.

In all heating systems, the condensate is returned to the boiler either by gravity or by mechanical means. In *gravity systems* the condensate is returned by gravity due to the static head of water in the return pipes or mains. The elevation of the boiler water line must be sufficiently below the lowest heating unit, steam pipe or dry return pipe to permit the return by gravity. The *water line difference* forming the *static head* must be sufficient to overcome the maximum pressure drop in the system, including the pressure drop due to the condensing effect of the radiation. When radiator and drip traps are used, as in two-pipe vapor systems, the static head must also exceed the operating pressure of the boiler. The pressure drop caused by condensing rate of the radiation is especially important during those portions of the operating periods when changing pressure conditions prevail, as for example, when the system is being initially filled with steam. In systems where the condensate is wasted to the sewer, no water line difference is required. However, the waste of condensate may introduce conditions which warrant the use of an appropriate mechanical return system. Whenever the conditions of a heating system are such that the condensate cannot gravitate to the boiler, it must be returned by some mechanical means.

In *mechanical systems* the condensate flows to a receiver by gravity and is then forced into the boiler against its pressure. In all instances the preferable practice is to provide for gravity flow even when a vacuum pump is used. The lowest parts of the supply side of the system must be kept sufficiently above the water line of the receiver to insure adequate drainage of water from the system.

There are three general types of mechanical return devices in common use, namely, (1) the mechanical return trap, (2) the condensate return pump, and (3) the vacuum return line pump.

PIPING FOR STEAM HEATING SYSTEMS

The functions of the piping system are the distribution of the steam, the return of the condensate and, in systems where no local air vents are provided, the removal of the air. The distribution of the steam should be rapid, uniform and without noise, and the release of air should be facilitated as much as possible, as an air bound system will not heat readily nor properly. In designing the piping arrangement it is desirable to maintain equivalent resistances in the supply and return piping to and

from a radiator. Arranging the piping so the total distance from the boiler to the radiation is the same as the return piping distance from the heating unit back to the boiler tends to obtain such a result. The condensate which occurs in steam piping as well as in radiators must be drained to prevent impeding the ready flow of the steam and air. The effect of back pressure in the returns and excessive revaporization, such as occurs where condensate is released from pressures considerably higher than the vacuum or pressure in the return, must be avoided.

It is important that steam piping systems distribute steam not only at full *design* load but during excess and partial loads. Usually the average winter steam demand is less than half of the demand at the *design* outside temperature. Moreover, in rapidly warming up a system even in moderate weather, the load on the steam main and returns may exceed the maximum operating load for severe weather due to the necessity of raising the temperature of the metal in the system to the steam temperature and the building to the *design* indoor temperature. Investigations of the

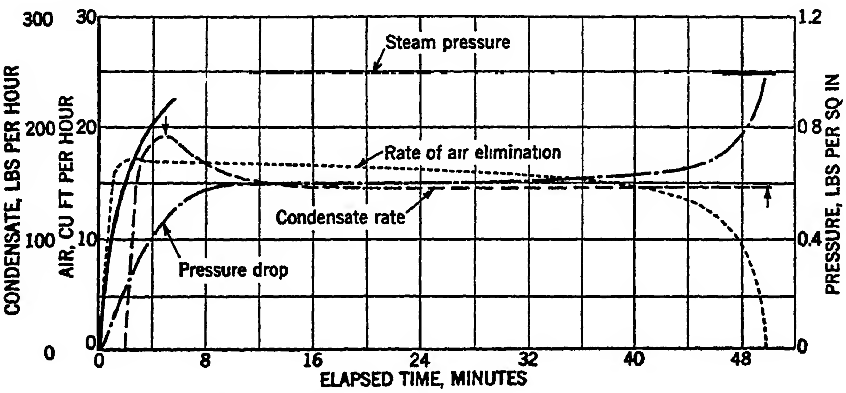


Fig. 1. Relation Between Elapsed Time, Steam Pressure, Condensate and Air Elimination Rates

return of condensate have revealed that as high as 143 per cent of the design condensation rate may exist under conditions of actual operation.

The piping design of a heating system is greatly influenced by its operating characteristics. Heating systems do not operate under constant conditions as they are continually changing due to variation in load. As the system is being filled with steam the pressures existing in various locations may be different from those which exist for appreciable periods at other locations although at equilibrium conditions the pressures are approximately the same. In designing piping it is of especial importance to arrange the system to preclude trouble caused by such pressure differences. The systems which readily release the air permit uniform pressures to be attained in much shorter time intervals than those which are sluggish. Results are given in Fig. 1 from investigations[1] to determine the rate of condensate and air return from a two-pipe gravity heating system. Variations in the steam pressure during the warming-up period when the rate of air elimination and condensation is high are clearly indicated in these curves.

It is evident that the condensate flow during the initial warming-up period reaches a peak which is greater than the constant condensing rate eventually reached when the pressure becomes uniform. Moreover,

the peak condensing rate is obtained when the system steam pressure is lower than that existing during a period of constant condensing rate. It will also be noted that the peak rate of air elimination does not coincide with the higher condensing rate.

STEAM FLOW

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction is in accordance with the general laws of gas flow and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship has been established by Babcock in the formula given at the top of Table 1. In Columns 1, 2, 3, and 4 of this table, the numerical values of the factors for different pressure losses, pipe diameters, steam densities and lengths of pipe have been worked out in convenient form so that the steam flowing in any pipe may be calculated by multiplying together the proper factors in each column as shown in the example at the bottom of the table.

PIPE SIZES

The determination of pipe sizes for a given load in steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and the end of the return system.
2. The maximum velocity of steam allowable for quiet and dependable operation of the system, taking into consideration the direction of condensate flow.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.
4. The direction of flow of the condensate, whether against or with the steam.

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered, such as initial pressure and pressure required at the end of the line, but it is most important that: (1) the total pressure drop does not exceed the initial gage pressure of the system and in actual practice it should never exceed one-half of the initial gage pressure; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the sub-atmospheric, which normally operate under controlled partial vacua, and orifice and vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; and (4) the rise in water due to pressure drop does not exceed the difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, or the dry return, and the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial gage pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing

Table 1. Flow of Steam in Pipes

P = loss in pressure in pounds per square inch.
D = inside diameter of pipe in inches.
L = length of pipe in feet.
d = weight of 1 cu ft of steam.
W = pounds of steam per hour.

$$W = 5220\sqrt{\frac{PdD^5}{\left(1+\frac{3.6}{D}\right)L}}$$

$$P = 0.0000000367\left(1+\frac{3.6}{D}\right)\frac{W^2L}{dD^5}$$

Pressure Loss in Ounces	Col. 1	Pipe Size		Internal Area of Pipe Sq Inches	Col. 2	Avg Steam Press. psig	Col. 3	Length of Pipe in Feet	Col. 4
	$5220\sqrt{\frac{P}{100}}$	Nominal	Actual Internal Diameter		$\sqrt{\frac{D^5}{1+\frac{3.6}{D}}}$		$\sqrt{d}$		$\sqrt{\frac{100}{L}}$
0.25	65.28	1	1.049	0.864	0.536	−1.0[a]	0.187	20	2.240
0.50	92.28	1¼	1.380	1.496	1.178	−0.5[a]	0.190	40	1.580
1.00	130.5	1½	1.610	2 036	1.828	0.0	0.193	60	1.290
2	184.6	2	2.067	3.356	3.710	0.3	0.195	80	1.120
3	226.0	2½	2.469	4.788	6.109	1.3	0.201	100	1.000
4	261.0	3	3.068	7.393	11.183	2.3	0.207	120	0.912
5	291.8	3½	3.548	9.887	16.705	5.3	0.223	140	0.841
6	319.7	4	4.026	12 730	23.631	10.3	0.248	160	0.793
7	345.3	4½	4.506	15.947	32.134	15.3	0.270	180	0.741
8	369.1	5	5.047	20.006	43.719	20.3	0.290	200	0.710
10	412.7	6	6.065	28.886	71.762	30.3	0.326	250	0.632
12	452.0	7	7.023	38.743	106.278	40.3	0.358	300	0.578
14	488.3	8	7.981	50.027	149.382	50.3	0.388	350	0.538
16	522.0	9	8.941	62.786	201.833	60 3	0.415	400	0.500
20	583.6	10	10.020	78.854	272.592	75.3	0.452	450	0.477
24	639.3	12	12.000	113.098	437.503	100.3	0.507	500	0.447
28	690.5	14	13.250	137.880	566.693	125.3	0 557	600	0.407
32	738.2	16	15.250	182.655	816.872	150.3	0.603	700	0.378
40	825.4					175.3	0.645	800	0.354
48	904.1					200 3	0.685	900	0.333
80	1167.2							1000	0.316
160	1650.7							1200	0.289
320	2334.5							1500	0.258
480	2859.1							2000	0.224

Column 1 × 2 × 3 × 4 = lb of steam per hour that will flow through a straight pipe for a given condition

Example 1: 1 oz drop — 2 in. pipe — 1.3 lb press — 100 ft equivalent length:

130.5 × 3.710 × 0 201 × 1 = 97.2 lb per hour.
97.2 × 4[b] = 388.8 sq ft equivalent radiation.

Table 1 does not allow for entrained water in low-pressure steam, condensation in covered pipe and roughness in commercial pipe as found in practice.

[a]Pounds per square inch gage = 2.04 in. Vacuum, Mercury Column.
[b]The factor 4 is the approximate equivalent in square feet of steam radiation of 1 lb of steam per hour.

factor is the velocity permissible without interfering with the condensate flow. A.S.H.V.E. Research Laboratory experiments limit this to the capacities given in Table 2 for horizontal pipes at varying grades.

Maximum Velocity

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensate present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of

TABLE 2. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES FOR STEAM AND CONDENSATE FLOWING IN OPPOSITE DIRECTIONS[a]

Pitch of Pipe in Inches per 10 Ft. Velocity in Ft per Sec.

PITCH OF PIPE	¼ IN.		½ IN.		1 IN.		1½ IN.		2 IN.		3 IN.		4 IN.		5 IN.	
Pipe Size Inches	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.
	Capacity Expressed in Square Feet E D R															
¾	25 0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209 3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263 5	20	299.5	23	325.5	25	346 5	27	371.5	28	388.4	29	401.1	30
	Capacity Expressed in Pounds per Hour															
¾	6.3	12	7.6	14	9.3	18	10.1	19	10.6	20	11.5	21	11.9	22	12.3	23
1	11.5	12	13.2	15	15.8	17	17.5	20	18.8	22	20.8	23	22.0	25	22.6	26
1¼	26.2	18	29.3	20	33.3	23	36.1	25	38.5	27	41.3	28	43.2	29	44.6	31
1½	35 7	18	39.8	21	45.3	23	49.1	25	52 3	27	56.0	28	58 7	30	60.7	31
2	59.0	19	65.9	20	74.9	23	81.4	25	86.6	27	92.4	28	97.1	29	100.3	30

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it runs horizontally, (3) the quantity of condensate flowing against the steam, and (4) freedom of the piping from water pockets which under certain conditions act as a restriction in pipe size.

Reaming Important

Three factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided. The second is the care used in reaming the ends of the pipe after cutting. The effect of both of these factors increases as the pipe size decreases. According to A.S.H.V.E. Research Laboratory tests, either of these factors may affect the capacity of a 1-in. pipe as much as 20 per cent. The third factor is the uniformity in grading the pipe line.

All of the capacity tables given in this chapter include a factor of safety. However, the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of

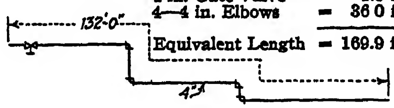
TABLE 3. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

Size of Pipe Inches	Length in Feet to be Added to Run				
	Standard Elbow	Side Outlet Tee	Gate Valve[a]	Globe Valve[a]	Angle Valve[a]
½	1.3	3	0.3	14	7
¾	1.8	4	0.4	18	10
1	2.2	5	0.5	23	12
1¼	3.0	6	0.6	29	15
1½	3.5	7	0.8	34	18
2	4.3	8	1.0	46	22
2½	5.0	11	1.1	54	27
3	6.5	13	1.4	66	34
3½	8	15	1.6	80	40
4	9	18	1.9	92	45
5	11	22	2.2	112	56
6	13	27	2.8	136	67
8	17	35	3.7	180	92
10	21	45	4.6	230	112
12	27	53	5.5	270	132
14	30	63	6.4	310	152

[a]Valve in full open position.

Example of length in feet of pipe to be added to actual length of run.

Measured Length	= 132.0 ft
4 in. Gate Valve	= 1.9 ft
4—4 in. Elbows	= 36 0 ft
Equivalent Length	= 169.9 ft



the same size of pipe. Table 3 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent length of run* as distinguished from the *actual length* of pipe in feet. The length of run is not usually known at the outset; hence it may be necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error and a more common and practical method is to assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

TABLES FOR PIPE SIZING FOR LOW PRESSURE SYSTEMS[2]

Tables 4, 5, and 6 are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent direct radiation (*abbreviated EDR*) per hour. The drops indicated are

TABLE 4. STEAM PIPE CAPACITIES FOR LOW PRESSURE SYSTEMS

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Pipe Size In.	Capacities of Steam Mains and Risers — Direction of Condensate Flow in Pipe Line — With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam Two-Pipe Only		Special Capacities for One-Pipe Systems Only		
	1/32 psi or 1/2 Oz Drop	1/24 psi or 3/4 Oz Drop	1/16 psi or 1 Oz Drop	1/8 psi or 2 Oz Drop	1/4 psi or 4 Oz Drop	1/2 psi or 8 Oz Drop	Vertical	Horizontal	Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[c]	*J*[b]	*K*	*L*[c]
Capacity Expressed in Square Feet E D R											
3/4	....	....	30	....	....	....	30	....	25	....	....
1	39	46	56	79	111	157	56	34	45	28	28
1 1/4	87	100	122	173	245	346	122	75	98	62	62
1 1/2	134	155	190	269	380	538	190	108	152	93	93
2	273	315	386	546	771	1,091	386	195	288	169	169
2 1/2	449	518	635	898	1,270	1,800	635	395	464	....	260
3	822	948	1,160	1,650	2,330	3,290	1,130	700	800	....	475
3 1/2	1,230	1,420	1,740	2,460	3,470	4,910	1,550	1,150	1,140	....	745
4	1,740	2,010	2,460	3,480	4,910	6,950	2,040	1,700	1,520	....	1,110
5	3,210	3,710	4,550	6,430	9,090	12,900	....	3,150	....	....	2,180
6	5,280	6,100	7,460	10,550	14,900	21,100	....	....	....	....	....
8	11,000	12,700	15,500	21,970	31,070	43,900	....	....	....	....	....
10	20,000	23,100	28,300	40,100	56,700	80,200	....	....	....	....	....
12	32,000	37,100	45,500	64,300	91,000	129,000	....	....	....	....	....
16	61,000	69,700	84,800	121,000	170,000	242,000	....	....	....	....	....
Capacity Expressed in Pounds per Hour											
3/4	....	....	8	....	....	....	8	....	6	....	7
1	10	12	14	20	28	40	14	9	11	7	7
1 1/4	22	25	31	43	61	87	31	19	20	16	16
1 1/2	34	39	48	67	95	135	48	27	38	23	23
2	68	79	97	137	193	273	97	49	72	42	42
2 1/2	112	130	159	225	318	449	159	99	116	....	65
3	206	237	291	411	581	822	282	175	200	....	119
3 1/2	307	355	434	614	869	1,230	387	288	286	....	186
4	435	503	614	869	1,230	1,740	511	425	380	....	278
5	806	928	1,140	1,610	2,270	3,210	....	788	....	....	545
6	1,320	1,520	1,870	2,640	3,730	5,280	....	....	....	....	....
8	2,750	3,170	3,880	5,490	7,770	11,000	....	....	....	....	....
10	5,010	5,790	7,090	10,000	14,200	20,000	....	....	....	....	....
12	8,040	9,290	11,400	16,100	22,700	32,200	....	....	....	....	....
16	15,100	17,400	21,200	30,300	42,400	60,500	....	....	....	....	....
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Un-dripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped

Note—All drops shown are in psi per 100 ft of equivalent run—based on pipe properly reamed.

[a]Do not use Column *H* for drops of 1/24 or 1/32 psi; substitute Column *C* or Column *B* as required.

[b]Do not use Column *J* for drop of 1/32 psi except on sizes 3 in. and over; below 3 in. substitute Column *B*

[c]On radiator runouts over 8 ft long increase one pipe size over that shown in Table 4.

Table 5. Return Pipe Capacities for Low Pressure Systems

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

Pipe Size Inches	Capacity of Return Mains and Risers																	
	1/32 Psi or 1/2 Oz Drop per 100 Ft			1/24 Psi or 2/3 Oz Drop per 100 Ft			1/16 Psi or 1 Oz Drop per 100 Ft			1/8 Psi or 2 Oz Drop per 100 Ft			1/4 Psi or 4 Oz Drop per 100 Ft			1/2 Psi or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
Mains																		
3/4	----	----	----	----	----	326	----	----	400	----	----	568	----	----	800	----	----	1,130
1	500	248	----	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	----	----	1,980
1 1/4	850	520	----	990	595	976	1,200	670	1,200	1,700	868	1,700	2,400	962	2,400	----	----	3,390
1 1/2	1,350	822	----	1,570	943	1,550	1,900	1,060	1,900	2,700	1,360	2,700	3,800	1,510	3,800	----	----	5,370
2	2,800	1,880	----	3,240	2,140	3,260	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	----	----	11,300
2 1/2	4,700	3,040	----	5,300	3,470	5,450	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	----	----	18,900
3	7,500	5,840	----	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,200	21,400	10,000	21,400	----	----	30,200
3 1/2	11,000	7,880	----	13,200	8,800	13,000	16,000	10,000	16,000	22,000	12,900	22,700	32,000	14,300	32,000	----	----	45,200
4	15,500	11,700	----	18,300	13,400	18,000	22,000	15,000	22,000	31,000	19,300	31,200	44,000	21,500	44,000	----	----	62,190
5	----	----	----	----	----	31,500	----	----	38,700	----	----	54,900	----	----	77,400	----	----	109,000
6	----	----	----	----	----	50,450	----	----	62,000	----	----	88,000	----	----	124,000	----	----	175,000
Risers																		
3/4	----	190	----	----	190	570	----	190	700	----	190	994	----	190	1,400	----	----	1,980
1	----	450	----	----	450	976	----	450	1,200	----	450	1,700	----	450	2,400	----	----	3,390
1 1/4	----	990	----	----	990	1,550	----	990	1,900	----	990	2,700	----	990	3,800	----	----	5,370
1 1/2	----	1,500	----	----	1,500	3,260	----	1,500	4,000	----	1,500	5,680	----	1,500	8,000	----	----	11,300
2	----	3,000	----	----	3,000	5,450	----	3,000	6,700	----	3,000	9,510	----	3,000	13,400	----	----	18,900
2 1/2	----	----	----	----	----	8,710	----	----	10,700	----	----	15,200	----	----	21,400	----	----	30,200
3	----	----	----	----	----	13,000	----	----	16,000	----	----	22,700	----	----	32,000	----	----	45,200
3 1/2	----	----	----	----	----	17,900	----	----	22,000	----	----	31,200	----	----	44,000	----	----	62,200
4	----	----	----	----	----	31,500	----	----	38,700	----	----	54,900	----	----	77,400	----	----	109,000
5	----	----	----	----	----	50,500	----	----	62,000	----	----	88,000	----	----	124,000	----	----	175,000

TABLE 6. RETURN PIPE CAPACITIES FOR LOW PRESSURE SYSTEMS

Capacity Expressed in Pounds per Hour

(Reference to this table will be made by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pipe Size Inches	CAPACITY OF RETURN MAINS AND RISERS																	
	1/32 Psi or 1/2 Oz Drop per 100 Ft			1/24 Psi or 2/3 Oz Drop per 100 Ft			1/16 Psi or 1 Oz Drop per 100 Ft			1/8 Psi or 2 Oz Drop per 100 Ft			1/4 Psi or 4 Oz Drop per 100 Ft			1/2 Psi or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
	Mains																	
3/4	...	...	...	...	...	42	...	...	100	...	...	142	...	...	200	...	...	283
1	125	62	...	145	71	143	175	80	175	250	103	249	350	115	350	...	...	494
1 1/4	213	130	...	248	149	244	300	168	300	425	217	426	600	241	600	...	...	848
1 1/2	338	206	...	393	236	388	475	265	475	675	340	674	950	378	950	...	...	1,340
2	700	470	...	810	535	815	1,000	575	1,000	1,400	740	1,420	2,000	825	2,000	...	...	2,830
2 1/2	1,180	760	...	1,580	868	1,360	1,680	950	1,680	2,350	1,230	2,380	3,350	1,360	3,350	...	...	4,730
3	1,880	1,460	...	2,130	1,560	2,180	2,680	1,750	2,680	3,750	2,250	3,800	5,350	2,500	5,350	...		7,560
3 1/2	2,750	1,970	...	3,300	2,200	3,250	4,000	2,500	4,000	5,500	3,230	5,680	8,000	3,580	8,000	...		11,300
4	3,880	2,930	...	4,580	3,350	4,500	5,500	3,750	5,500	7,750	4,830	7,810	11,000	5,380	11,000	...	...	15,500
5	...	...	...	...	...	7,880	...	...	9,680	...	...	13,700	...	...	19,400	...		27,300
6	...	...	...	...	...	12,600	...	...	15,500	...	...	22,000	...	...	31,000	...	...	43,800
	Risers																	
3/4	...	48	...	...	48	143		48	175	...	48	249	...	48	350	...	...	494
1	...	113	...	...	113	244	...	113	300	...	113	426	...	113	600	...	...	848
1 1/4	...	248	...	...	248	388	...	248	475	...	248	674	...	248	950	...	...	1,340
1 1/2	...	375	...	...	375	815	...	375	1,000	...	375	1,420	...	375	2,000	...	...	2,830
2	...	750	...	...	750	1,360	...	750	1,680	...	750	2,380	...	750	3,350	...	...	4,730
2 1/2	...	...	...	...	...	2,180	...	...	2,680	...	...	3,800	...	...	5,350	...	...	7,560
3	...	...	...	...	...	3,250	...	...	4,000	...		5,680	...	...	8,000	...		11,300
3 1/2	...	...	...	...	...	4,480		...	5,500	...		7,810	...	...	11,000	...		15,500
4	...	...	...	...	...	7,880		...	9,680	...	...	13,700	...	...	19,400	...		27,300
5	...	...	...	...	...	12,600	...	...	15,500	...	...	22,000		...	31,000	...	...	43,800

drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed and without unusual or noticeable defects.

Table 4 may be used for sizing piping for steam heating systems by pre-determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used where the steam and condensate flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensate flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve and vertical connection sizes, and radiator and runout sizes, all of which are based on the critical velocities of the steam to permit the counter flow of condensate without noise.

Return piping may be sized with the aid of Tables 5 and 6 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 4. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

Example 2. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft and the initial pressure is not to be over 2-psi gage?

Solution. It will be assumed, if the measured length of the longest run is 500 ft, that when the allowance for fittings is added the equivalent length of run will not exceed 1,000 ft. Then, with the pressure drop not over one half of the initial pressure, the drop could be 1 psi or less. With a pressure drop of 1 psi and a length of run of 1,000 ft, the drop per 100 ft would be $\frac{1}{10}$ psi, while if the total drop were $\frac{1}{2}$ psi, the drop per 100 ft would be $\frac{1}{20}$ psi. In the first instance the pipe could be sized according to Column *D* for $\frac{1}{16}$ psi per 100 ft, and in the second case, the pipe could be sized according to Column *C* for $\frac{1}{24}$ psi. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used and the lines resized. Ordinarily resizing will be unnecessary.

PIPE SIZING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized in a manner similar to radiators, but the equivalent direct radiation must be ascertained for each row of heating unit stacks and then must be divided into the number of stacks constituting that row and into the number of connections to each stack.

$$EDR = \frac{Q \times 60 \times (t_l - t_e)}{55.2 \times 240} = \frac{Q \times (t_l - t_e)}{220.8} \qquad (1)$$

where

EDR = equivalent direct radiation, square feet.

Q = volume of air, cubic feet per minute.

t_e = the temperature of the air entering the row of heating units under consideration, Fahrenheit degrees.

t_l = the temperature of the air leaving the row of heating units under consideration, Fahrenheit degrees.

60 = the number of minutes in one hour.

55.2 = the number of cubic feet of air heated 1 Fahrenheit degree by 1 Btu.

240 = Btu equivalent of 1 sq ft of EDR.

Example 3. Assume that a 3-row heating unit shown in this chapter in Fig. 38 is handling 50,000 cfm of air and that the rise in the first row is from 0 to 40 F, in the second row from 40 to 65 F, and in the third row from 65 to 80 F. What is the load in EDR on each supply and return connection?

Solution. For row 1,

$$R = \frac{50{,}000 \times (40 - 0)}{220.8} = 9058 \text{ sq ft.}$$

For row 2,

$$R = \frac{50{,}000 \times (65 - 40)}{220.8} = 5661 \text{ sq ft.}$$

For row 3,

$$R = \frac{50{,}000 \times (80 - 65)}{220.8} = 3397 \text{ sq ft.}$$

Each row of heating units consists of four stacks and each stack has two connections so that the load on each stack and each connection of the stack is as follows:

Row	Total Load (EDR)	Stack Load[a] (EDR)	Connection Load[b] (EDR)
1	9058	2265	2265 or 1132
2	5661	1415	1415 or 708
3	3397	849	849 or 425

[a]One quarter of total row load.
[b]One half of stack load if two steam connections are made; otherwise, same as stack load.

The pipe sizes would then be based on the length of the run and the pressure drop desired, as in the case of radiators. It generally is considered desirable to place the indirect heating units on a separate system and not on supply or return lines connected to the general heating system.

Most manufacturers of heating coils now publish condensing capacities of their coils for all common operating conditions in pounds of steam per hour per square foot of face area (or free area for some designs). These rates may be used in connection with Tables 4 and 6 to size the piping.

GRAVITY ONE-PIPE AIR-VENT SYSTEM

This system is the most common of all methods of steam heating, especially for small size installations, due largely to its low cost and simplicity.

The downward pitch of a one-pipe air-vent system is indicated in Fig. 2. Low points and ends of steam mains pitched down from the boiler should be dripped. All drips should be sealed below water line before connecting together. In the risers and radiator connections, steam and condensate flow in opposite directions. In long steam mains it flows in the same direction as the steam and is removed from the main through the drip. Short mains may be arranged for the condensate to flow in a direction opposite the steam by sizing them so the critical velocity is not exceeded. It is customary to drip the heel of each riser in buildings of several stories to avoid counter-flow of the steam and condensate in the riser runout. In buildings of one or two stories the condensate is returned to the steam main instead of being dripped. Both types of risers are shown in Fig. 2, and riser connections are shown in Figs. 3 and 4. A typical overhead down-feed system is illustrated in Fig. 5. While wet

return mains need not be pitched toward the boiler to maintain steam circulation, they should be pitched for drainage.

To improve steam circulation in one-pipe systems quick vent air valves should be provided at the ends and at intermediate points where the steam main is brought to a higher elevation. It is desirable to install the air-vent valves about a foot ahead of the drips, as indicated in Fig. 2, to prevent possible damage to their mechanisms by water.

The radiator valves may be the angle-globe, offset-corner pattern or gate type. Straight-globe and straight-corner types should not be used since the raised valve seat would interfere with the flow of condensate through the valve. Graduated valves cannot be used since the steam valves on this system must be fully open or fully closed to prevent the radiators filling with water and creating a dangerous water line condition.

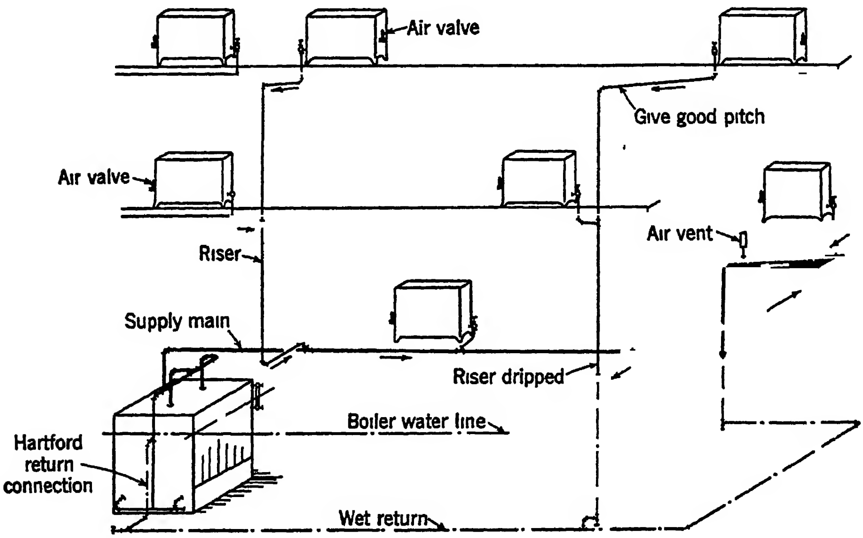


Fig. 2. Typical Up-Feed Gravity One-Pipe Air-Vent System

With a one-pipe system the heat cannot be modulated at the radiator, the steam being either *all on* or *all off*. Systems and devices are available which make it possible to obtain a partial modulating effect from one-pipe heating systems.

It is important to keep the lowest points of the steam mains and heating units sufficiently above the water line of the boiler to prevent flooding. The minimum water line difference depends on the initial steam pressure and piping pressure drop plus a safety factor for heating up.

Referring to Fig. 6 it will be noted that the boiler and wet return form a U-shaped container, with the boiler steam pressure on the top of the water at one end and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, *i.e.*, the friction and resistance to the flow of steam in passing from the boiler to the far end of the main and the pressure reduction in consequence of the condensation occurring in the system.

The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise far enough to produce a flow through the return pipe and overcome the resistance of check valves if installed.

If a one-pipe steam system is designed, for example, for a total pressure drop of $\frac{1}{8}$ psi, and utilizes a Hartford return connection instead of a check valve on the return, the rise in the water level at the far end of the return due to the difference in steam pressure would be $\frac{1}{8}$ of 28 in. (28 in.

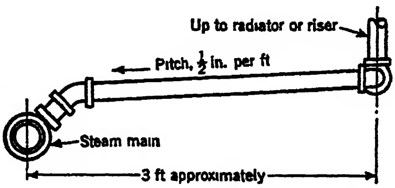


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

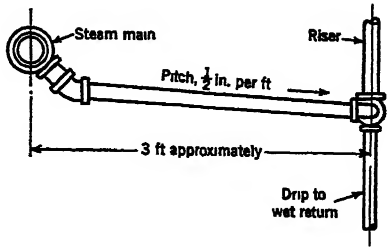


FIG. 4. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

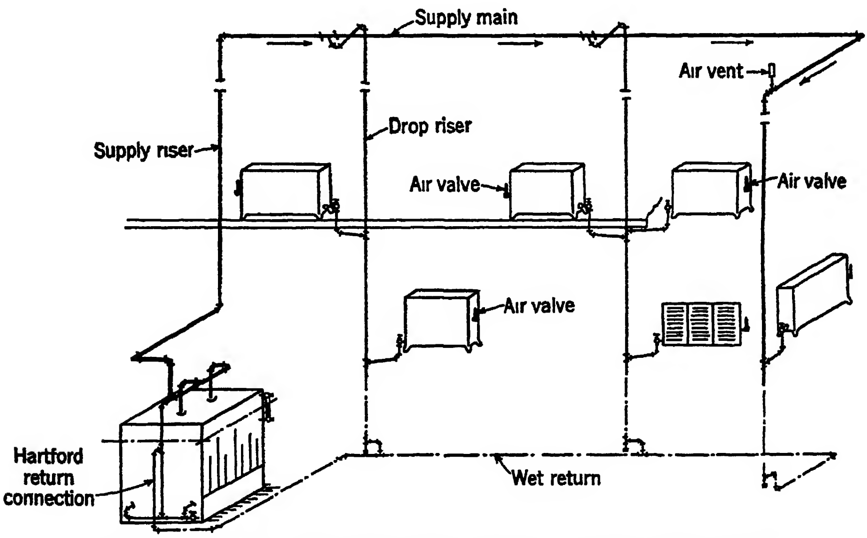


FIG. 5. TYPICAL DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

head being equal to one pound per square inch), or $3\frac{1}{2}$ in. Adding 3 in. to overcome the resistance of the return main and 6 in. as a factor of safety for heating up gives $12\frac{1}{2}$ in. as the distance the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of $\frac{1}{2}$ psi, and with a check in the return, would require $\frac{1}{2}$ of 28 in., or 14 in. for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

Gravity one-pipe air-vent systems in which the equivalent length of run does not exceed 200 ft should be sized by means of Tables 4, 5 and 6 as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use 1/16-psi drop (Column *D*).

2. *Where the riser runouts are not dripped* and the steam and condensate flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.

3. *For up-feed steam risers* carrying condensate back from the radiators, use Column *J*.

4. *For down-feed systems the main risers* of which do not carry any radiator condensate use Column *H*.

5. *For the radiator valve size and the stub connection,* use Column *K*.

6. For the *dry return main,* use Column *U*.

7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over 1/4 psi. The return piping sizes should corres-

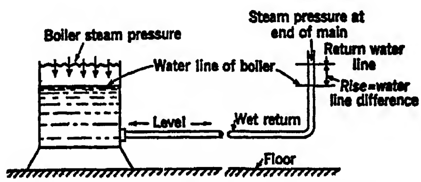


Fig. 6. Difference in Steam Pressure on Water in Boiler and at End of Steam Main

pond with the drop used on the steam side of the system. Thus, where 1/24-psi drop is being used, the steam main and dripped runouts would be sized from Column *C*; radiator runouts and undripped riser runouts from Column *L*; up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of 1/24 psi); the dry return from Column *R*; and the wet return from Column *Q*.

With a 1/32-psi drop the sizing would be the same as for 1/24 psi except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should not be less than 1/4 in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than 1/2 in. per foot. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3. In general, it is not desirable to have a main less than 2 in. The diameter of the far end of the supply main should not be less than half its diameter at its largest part.

4. Supply mains, runouts to risers, or risers, should be dripped where necessary.

5. Where supply mains are decreased in size they should be dripped, or be provided with eccentric couplings, flush on bottom.

Example 4. Size the one-pipe gravity steam system shown in Fig. 7 assuming that this is all there is to the system or that the riser and main shown involve the longest run on the system.

Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of 1/4 psi

TABLE 7. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG. 7

Part of System	Section of Pipe	Radiation Supplied EDR Sq Ft	Theoretical Pipe Size (Inches)	Practical Pipe Size (Inches)
Branches to radiators	......	100	2	2
Branches to radiators	......	50	1¼	1¼
Riser	*a* to *b*	200	2	2
Riser	*b* to *c*	300	2½	2½
Riser	*c* to *d*	400	2½	2½
Riser	*d* to *e*	500	3	3
Riser	*e* to *f*	600	3	3
Runout to riser	*f* to *g*	600	3½	3½
Supply main	*g* to *h*	600	3	3
Branch to supply main	*h* to *j*	600	2½	3
Dry return main	*f* to *k*	600	1¼	2
Wet return main	*k* to *m*	600	1	2
Wet return main	*m* to *n*	600	1	2
Wet return main	*n* to *p*	600	1	2

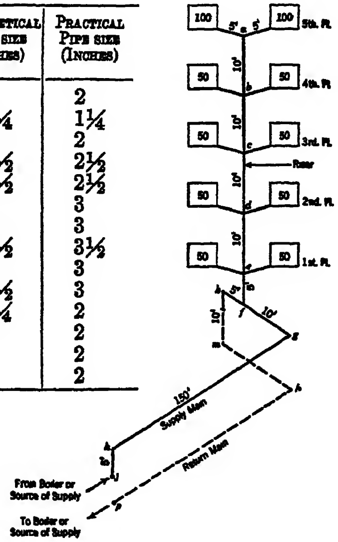


FIG. 7. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

the drop per 100 ft will be slightly less than $\frac{1}{16}$ psi. It would be well in this case to use $\frac{1}{24}$ psi, and this would result in the theoretical sizes indicated in Table 7. These theoretical sizes, however, should be modified by not using a wet return less than 2 in. while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in. if the wet return is made 2 in.

ONE-PIPE VAPOR SYSTEM

The one-pipe vapor system operates under pressures at or near atmospheric and returns its condensate to the boiler by gravity. In this

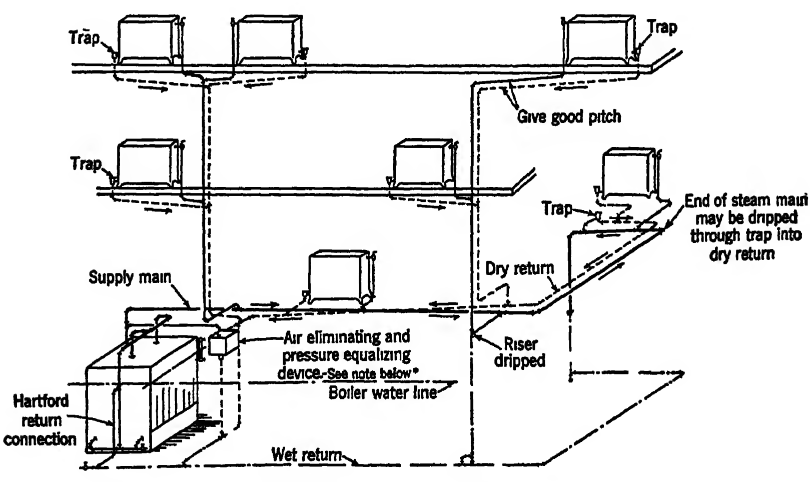


FIG. 8. TYPICAL UP-FEED SYSTEM WITH AUTOMATIC RETURN TRAP*

*Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

system the automatic air valves are of special design to permit the ready release of air and prevent its ready return after it is expelled. The steam radiator valves are a type which, when opened, give a free and unobstructed passageway for water. The piping is the same as for the one-pipe gravity system but sized so as to permit operation at a few ounces pressure.

TWO-PIPE VAPOR SYSTEM

A two-pipe up-feed vapor system using separate supply and return pipes is shown in Fig. 8. The radiators discharge their condensate through thermostatic traps to the dry return pipe. These systems operate at a few ounces pressure and above, but those with mechanical condensate return devices may operate at pressures upward of 10 psi. The simplest

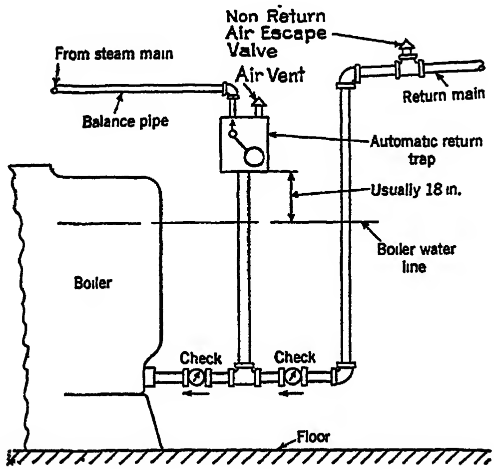


Fig. 9. Typical Connections for Automatic Return Trap

method of venting the system consists of a ¾-in. pipe with a check valve opening outward. Most systems employ various forms of vent valves, designed to allow the air to readily pass out of the system and to prevent its return. These systems permit control of the heat in the radiator by varying the opening of the graduated radiator valves. The boiler pressure is maintained at substantially constant pressure slightly above atmospheric pressure.

These systems may be classified as (1) *closed systems*, consisting of those which have a device to prevent the return of air after it has once been expelled from the system, and which can operate at both super and subatmospheric pressures for a period of four to eight hours depending upon the tightness of the system and rate of firing, and (2) *open systems*, comprising those which have the return line constantly open to the atmosphere without a check or other means to prevent the return of air. The open systems are not so popular because they have the disadvantage of not holding heat when the rate of steam generation is diminishing.

Closed systems should preferably be equipped with an automatic return trap to prevent water from backing out of the boiler. In installing the return trap a check valve is inserted in the return main at a point near the boiler and a vertical pipe is run up into the bottom of the return trap,

which is usually located with the bottom about 18 in. above the boiler water line. Some traps are constructed so that they will operate when they are installed with their bottom as close as 8 in. above the boiler water line. On the other side of this connection a second check valve is installed in the main return just before it enters the boiler. Fig. 9 shows a typical connection for an automatic return trap.

Down-Feed Two-Pipe Vapor System

In the down-feed two-pipe vapor system the steam is carried to the top of the building, the top of the vertical riser constituting the high point of the system, and the horizontal supply main is sloped down from this location to the far ends of each branch. The runouts are taken off the main from the bottom or at a 45-deg angle downward, and sloped toward the drops. Thus each runout from the main forms a drip and no accumulation of water is carried down any one drop.

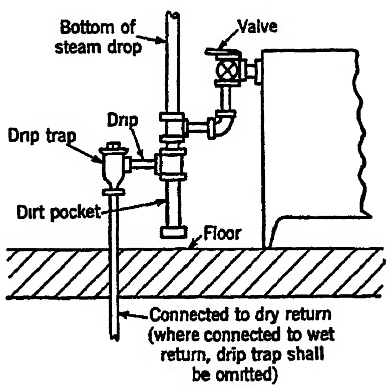


FIG. 10. DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

The steam drops are carried down through the building with suitable reductions as the various radiator connections are taken off until the lowest radiator runout is reached. If the drop is only two or three stories high, the portion feeding the bottom radiator should be increased one pipe size to provide for draining the riser, and if the drop is over three stories high it is well to increase the portion feeding the two lowest radiators one or two pipe sizes, especially if the two lowest radiators are small and the normal size of drop required is 1 in. or less. The bottom of each steam drop should terminate with a dirt pocket and be dripped as shown in Fig. 10. The returns on a down-feed vapor system are the same as on an up-feed system. The runouts to the radiators and the radiator connections of the down-feed system are the same as those for the up-feed system already described.

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensate return to the boiler by gravity, (2) to obtain a more uniform distribution of steam throughout the system, especially when it is desirable to carry a moderate or low

fire, and (3) because with large variations in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped should be sized from Column *D*, Table 4, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Tables 5 and 6, Column *U*, (lower portion) and the mains from Column *U* (upper portion). It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side except where the flow in the riser is concerned.

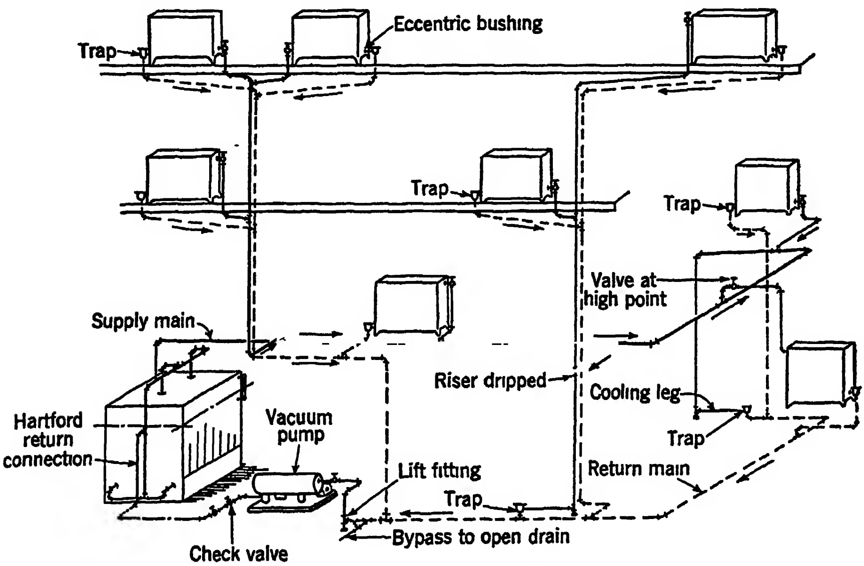


FIG. 11. TYPICAL UP-FEED VACUUM PUMP SYSTEM

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* for small systems correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed ⅛ psi to ¼ psi, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over ⅛ psi divided by 4, or 1/32 psi. In this case the steam mains would be sized from Column *B*, the radiator and undripped riser runouts from Column *I*; the risers from Column *B*, because Column *H* gives a drop in excess of 1/32 psi. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over 1/32 psi. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1. Pitch of mains should not be less than ¼ in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. per foot. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3. In general it is not desirable to have a supply main smaller than 2 in.

4. When necessary, supply main, supply risers, or runouts to supply risers should be dripped separately into a wet return, or may be connected into the dry return through a thermostatic drip trap.

VACUUM SYSTEMS

In the vacuum system, a vacuum is maintained in the return line practically at all times. The pump is usually controlled by a vacuum regulator which operates the pump to maintain the vacuum within limits and operates in response to a pressure difference between the atmosphere and the return to control the vacuum in the return main. The source of steam supply may be a low pressure boiler as shown in Fig. 11, or a high pressure line through a pressure reducing valve. The piping and other details are the same as for the vapor systems.

The return risers are connected in the basement into a common return main which slopes downward toward the vacuum pump. The vacuum pump withdraws the air and water from the system, separates the air from the water and expels it to atmosphere and pumps the water back to the boiler, or other receiver, which may be a feed-water heater or hot well. It is essential that no connection be made from the supply side to the return side at any point except through a trap. The desirable practice demands a return flowing to the vacuum pump by an uninterrupted downward slope. In some instances local conditions make it necessary to drop the return below the level of the vacuum pump inlet, before the pump can be reached. In such an event one of the advantages of the vacuum system is the ability to raise the condensate to a considerable height by the suction of the vacuum pump by means of a lift connection or fitting inserted in the return. The height the condensate can be raised depends on the amount of vacuum maintained. It is preferable to limit lift connections to a single lift at the vacuum pump. A still more preferable arrangement is the use of an accumulator tank, or receiver tank, with a float control for the pump, at the low point of the return main located adjacent to the vacuum pump.

When the vertical lift is considerable, several lift fittings should be used in steps as shown in Fig. 12. This permits a given lift to be secured with a somewhat lower vacuum than where the vertical distance is served by a single lift. Where several lifts are present in a given system at different locations, the lifting cannot occur until the entire system is filled with steam. A lift connection for location close to the pump, where the size may be above the commercial stock sizes, is shown in Fig. 13. It is desirable that means be provided for manually draining the low point of the lift fittings to eliminate danger of freezing.

Down-Feed Vacuum System

The piping arrangement for the down-feed vacuum system is similar on the supply side to the down-feed vapor system in that it has similar runouts, radiator valves, drips on the bottom of the steam drops, and enlargement of the drops for the lower radiator connections. The return

side of the system is exactly the same as the up-feed system except that the steam riser drips at the bottom are connected into the return line through thermostatic traps. It is preferable to take the runouts for the risers from the bottom or at a 45-deg angle down from the steam main so that they may serve as steam main drips. When this is done it is practical to run the steam main level, if a runout is located at every change in pipe size, or if eccentric fittings are used (Fig. 14). A slight pitch in the steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 15.

Vacuum, atmospheric, sub-atmospheric and orifice systems are usually employed in large installations and have total drops varying from ¼ to ½ psi. Systems where the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop, while systems over 200 ft equivalent length of run more frequently are designed for the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of ½ psi divided by 12, or 1/24 psi. In this case, the steam main would be sized from Column *C*, Table 4, and the risers also from Column

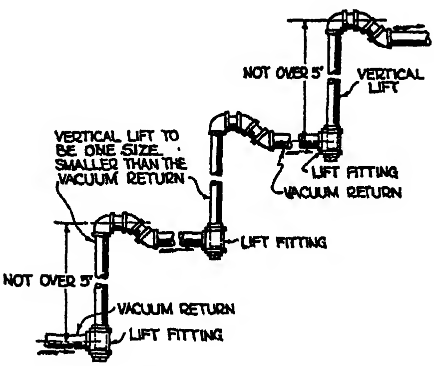


Fig. 12. Method of Making Lifts on Vacuum Systems when Distance is Over 5 ft

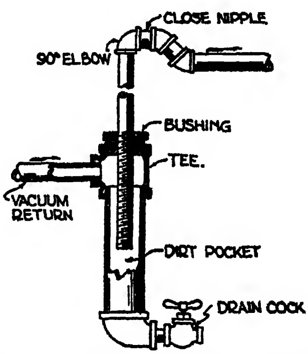


Fig. 13. Detail of Main Return Lift at Vacuum Pump

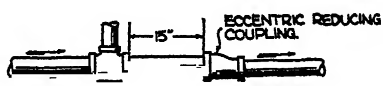


Fig. 14. Method of Changing Size of Steam Main when Runouts are Taken from Top

C (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of 1/24 psi). Riser runouts, if dripped, would use Column *C* but if undripped would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*, Tables 5 and 6; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed ⅛ psi drop per 100 ft of equivalent run nor to exceed 1 psi total pressure drop in any system.

2. Pitch of mains should not be less than ¼ in. in 10 ft.

3. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. per foot. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

4. In general it is not considered desirable to have a supply main smaller than 2 in.

5. When necessary, the supply main, supply riser, or runout to a supply riser should be dripped separately through a trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a trap to prevent the steam from entering the return line.

6. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in this chapter.

7. No lifts can be used in orifice and atmospheric systems. In sub-atmospheric systems the lift must be at the vacuum pump.

SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems but, in contrast, provide control of building temperature by variation of the heat

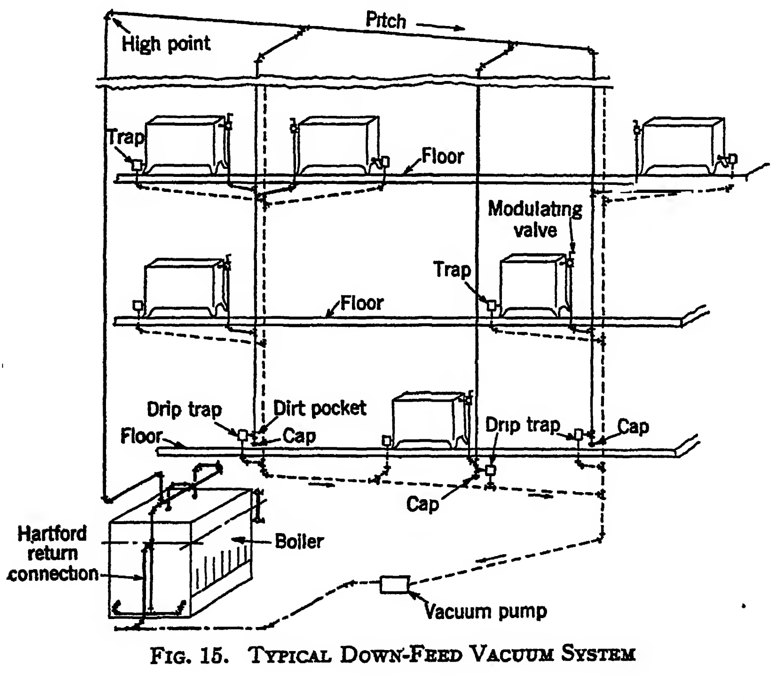


FIG. 15. TYPICAL DOWN-FEED VACUUM SYSTEM

output from the radiators. The radiator heat emission is controlled by varying the pressure, temperature and specific volume of steam in circulation. These systems differ from the ordinary vacuum system in that they maintain a controllable partial vacuum on both the supply and return sides of the system, instead of only on the return side. In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, atmospheric pressure or higher exists in the steam supply piping and radiators only during severe weather. Under average winter temperature the steam is under partial vacuum which in mild weather may reach as high as 25 in. Hg, after which further reduction in heat output is obtained by restricting the quantity of steam.

The rate of steam supply is controlled by a valve in the steam main or by thermostatically controlling the rate of steam production in the boiler. The control valve may be of the automatic *modulating* or *floating* type governed thermostatically from selected control points in the building, or it may be a special pressure reducing valve which will maintain the desired sub-atmospheric pressures by continuous flow into the heating main. All radiator supply valves have incorporated adjustable orifices or are equipped with regulating orifice plates. The sizes of orifices used are larger than for other types of orifice systems because for equal radiator sizes the volume flowing is larger. These orifices are omitted on some systems, depending upon the type of control. Radiator traps and drips are designed to operate at any pressure from 15 lb gage to 26 in. Hg. A vacuum pump capable of operating at high vacuum is preferable to promote accuracy in the distribution of steam throughout the system, particularly in mild weather. This vacuum is partially self-induced by the condensation of the steam in the system under conditions of restricted supply for reduction of the radiator heat emission.

The returns must grade downward constantly and uninterruptedly from the radiator return outlets to the inlet of the receiver of the vacuum pump. One radical difference between this and the ordinary vacuum system is that no lifts should be made in the return line, except at the vacuum pump. The receivers are placed at a lower level than the pump and equipped with float control so that the pump may operate as a return pump under night conditions. The system may be operated in the same manner as the ordinary vacuum system when desired.

Steam for heating domestic hot water should be taken from the boiler header back of the control valve so that pressures sufficiently high for heating the water may be maintained on the heater. The sub-atmospheric method of heating can be used for the heating coils of ventilating and air conditioning systems. The flexible control of heat output secured by this method materially reduces the required size of by-pass around the heaters. Some applications of sub-atmospheric systems are proprietary.

ORIFICE SYSTEMS

Orifice steam heating systems may have piping arrangements identical with vacuum systems. Some of these omit the radiator thermostatic traps but use thermostatic or combination float and thermostatic traps on all drip points. A return condensate pump with receiver vented to atmosphere, a return line vacuum pump, or a return trap, is generally used to return the condensate to the boiler or place of similar disposition, such as a feed-water heater or hot well. The heat emission from the radiators is controlled by varying the pressure differential maintained.

The principle on which these systems operate is based on the fact that the steam flow through an orifice will vary when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent of the absolute pressure on the inlet side, no further increase in flow will be obtained as a result of the increased pressure difference. If an orifice is so designed in size as to exactly fill a radiator with 2 psi gage on one side and ¼ psi gage on the other, the absolute pressure relation is:

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 0.90 \text{ or } 90 \text{ per cent.}$$